The EDITOR'S PAGE

Heating · Piping *and* Air Conditioning

THIS MONTH—

Maximum fuel-burning rate is an important item to consider in selecting a heating boiler. Inability to burn sufficient coal, oil or gas (as the case may be) will prolong the warming-up period and may result in failure to carry the designed radiation load.

Sabin Crocker's article "Selecting the Right Size Heating Boiler" in this number is devoted to a discussion of the fuel-burning rate; a number of tables are included which will be found useful when selecting a heating boiler.

While methods of testing high-pressure steam lines are not definitely standardized, much progress along these lines has been made. It is John J. Harman's thought that as the working pressures and temperatures in common use become stabilized, standardization of testing will be accomplished.

Mr. Harman has prepared an interesting and valuable paper in "Testing High-Pressure Steam Piping"—our "lead-off" this month.

Mr. Harman's article is confined mainly to flanged joints. A logarithmic chart which shows the turning moment required to produce various initial bolt stresses in bolts of different sizes should be of especial interest; while it is based on a theoretical formula, a sufficient number of points have been checked against actual experimental data to give assurance that the chart closely approximates average field conditions.

Piping engineers will perhaps recall—in connection with the subject of testing—the paper "How Welds Are Tested," which was published in last December's issue.

L. A. Harding contributes some comments on determining total, static and velocity pressure to the "Open for Discussion" department this month, and illustrates an example with a pressure loss diagram.

In "Calculations for Drying" Malcolm Tomlinson presents four charts which are of considerable utility in solving problems concerning drying at high temperatures. Mr. Tomlinson illustrates their use in typical examples given in his paper, which is a not-to-be-missed part of this issue.

Developments—both technical and commercial—have been numerous lately in the air conditioning of railway cars. At the present time,

several air-conditioned cars are being built for various roads and will go in service this summer. Air conditioning has proved its worth as an inducement to the public in the contest for traffic.

In an article in this issue several of the methods used in conditioning air for the comfort of passengers are described and the technical factors that must be considered in conditioning cars are outlined. It is a subject which is particularly timely.

NEXT MONTH—

In the article by Robert L. Fitzgerald, published in March, the new mile-long underground line through which the Indianapolis Power & Light Co. supplies steam for process to the Eli Lilly plant was described. Next month, William A. Hanley, chief engineer of Eli Lilly and Company, will tell how the steam is utilized, metered and paid for. Mr. Hanley quotes at length from the contract between the utility and his concern in an article which—because of the economic importance of this form of cooperation between a plant and a utility—no industrial engineer will want to miss. Why this method of obtaining process steam to meet its increasing needs was used by Eli Lilly and Company is treated in detail.

John H. Ruckman writes on preventing air pollution in the plant in the next issue, supporting his points with actual examples. A classification of objectionable materials is included.

A large and relatively important class of buildings (churches, theaters, auditoriums, gymnasiums, etc.) are heated intermittently, yet a search of current literature reveals little published information on this topic. As a result of experience with some 600 buildings used intermittently, John M. Robertson contributes an informative article to the May HEATING, PIPING AND AIR CONDITIONING. In it, he describes the methods of making tests to determine the total heat required to raise the temperature of a known mass of building materials, and shows some results graphically.

Illustration is an Honest Exaggeration of a 10x10 Weldolet

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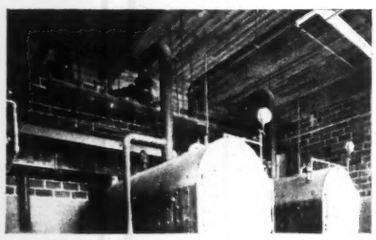

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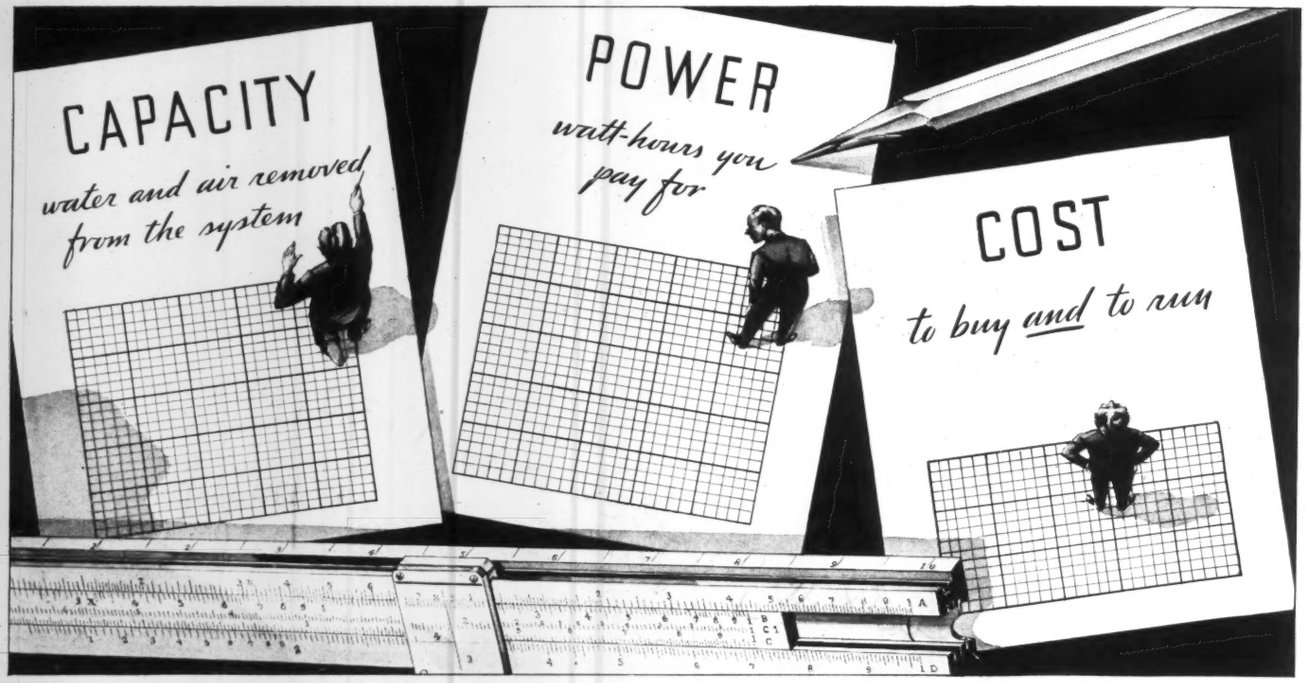

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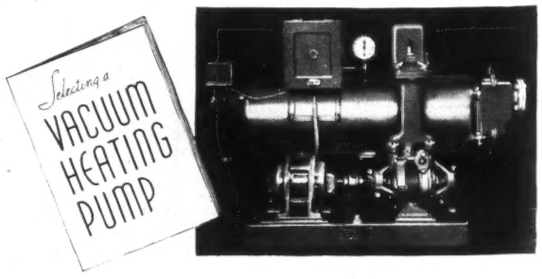

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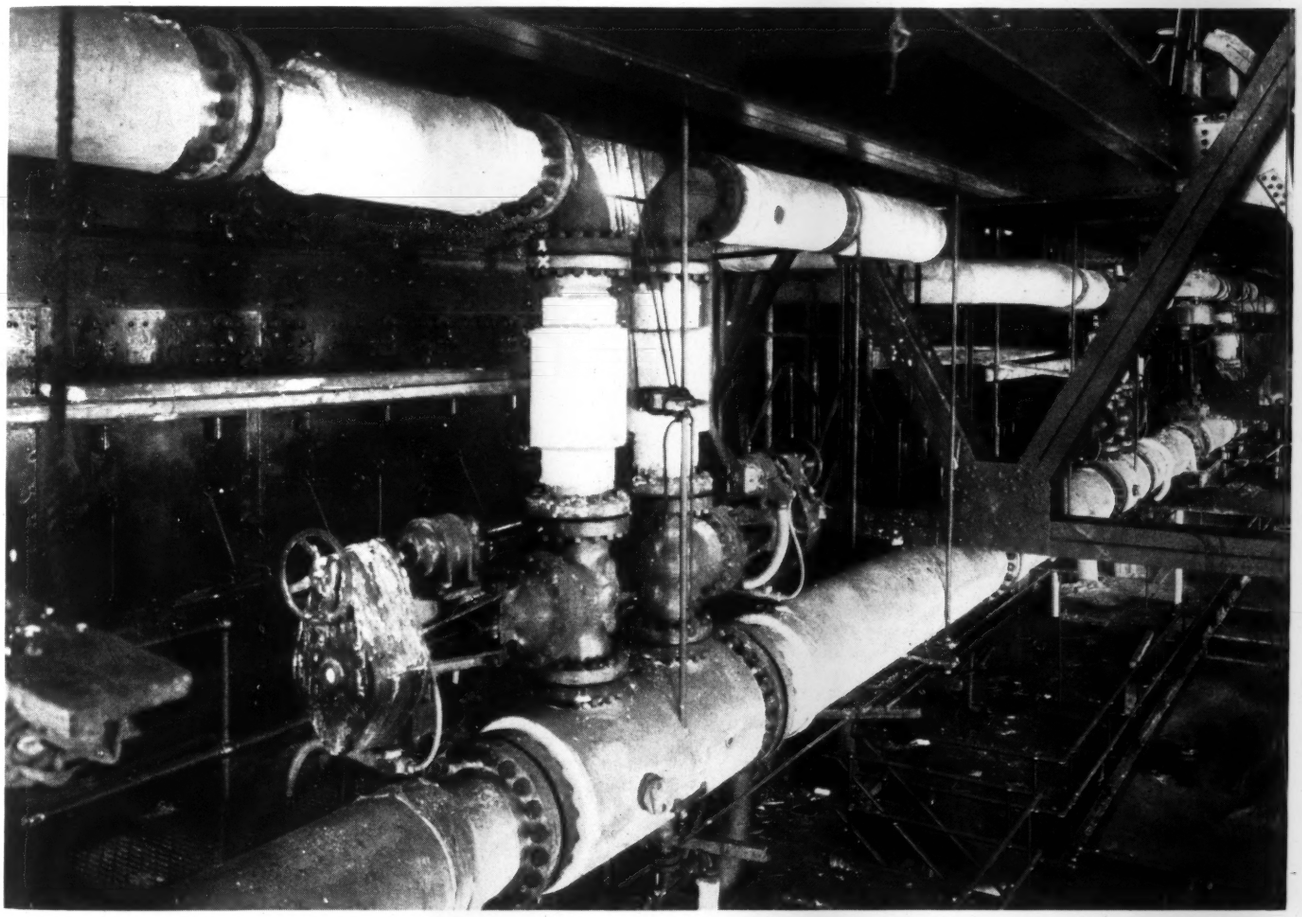


Boiler House Piping for Steam at 400 Lb. and 750 F, Before Installation of Covering Was Completed

Testing High-Pressure Steam Piping

By John J. Harman*

THE term *high-pressure steam line* implies high temperature, because high-pressure steam when carried in pipes is practically always in a highly superheated condition. Pressures and temperatures from 250 to 1,800 lb. per sq. in. and from 450 to 1,000 F introduce difficulties in the making and assembling of pipe joints that will be permanently tight. The penetrating power and disruptive force of the high pressure, supplemented by the dilatation due to high temperature and the working of the joints caused by expansion and contraction of the pipe line, tend to open the joints and cause leaks, which unfortunately develop in some cases after the system has been in operation for a considerable period of time. In order to have permanently tight joints in this class of piping, it consequently is neces-

*Assistant chief engineer, Walworth Company, Boston, Mass.

sary to pay particular attention to the lay-out of the piping system and to the design of the joints, as well as to use special care in the erection of the piping and assembling and testing of the joints. It therefore appears worth while to discuss these briefly, for no amount of attention given to the assembling and testing of joints can correct fundamental mistakes in design.

Care in Lay-out Needed

Piping systems operating at temperatures of from 700 to 1,000 F expand a great deal more than low temperature systems, not only because the temperature range is larger but also because the coefficient of expansion is greater at the higher temperatures. The lay-out therefore should provide sufficient flexibility to prevent setting up excessive expansion stresses[1], hangers and supports should be designed to permit free movement and should be close enough together to eliminate excessive bending stresses due to the weight of the pipe filled with water together with the weight of the fittings, valves and pipe covering. Stiff cross-connections and other design features that would introduce severe bending stresses should also be avoided.

[1] For method of computing stresses caused by pipe line expansion see "Piping Handbook," Walker and Crocker, McGraw-Hill Book Co.

Standards for Flanged Joints

Review of Standard Dimensions for Flanged Joints . . . . Flange Material . . . . Bolts . . . . Threads . . . . Nuts . . . . Finish of Flange Faces

Standard dimensions for flanged joints have been established in *Tentative American Standard for Steel Pipe Flanges and Flanged Fittings B16e* for steam working pressures of 250, 400, 600, 900, and 1,350 lb. at a temperature of 750 F. This pressure and temperature range was thought to be adequate for the probable requirements at the time it was adopted in 1927, but almost immediately the upper limits of both temperature and pressure were exceeded, both in the oil industry and in the steam power plant field. Some changes have been made in the standard since its original adoption and it is probable that the 250-lb. standard will be changed in the near future to a 300-lb. standard and the 1,350-lb. standard to a 1,500-lb. standard in order to match more closely the actual service conditions that have developed. Some changes in shell thicknesses are contemplated in both standards, and some changes in flanges, bolting, and face to face dimensions in the 1,350-lb. standard for sizes above 4 in. For pressures above 1,500 lb. and temperatures above 750 F there are at present no generally accepted standard dimensions for flanged joints and in this zone the designs that have been used are not uniform and reflect the individual ideas of the designers responsible for the particular installations.

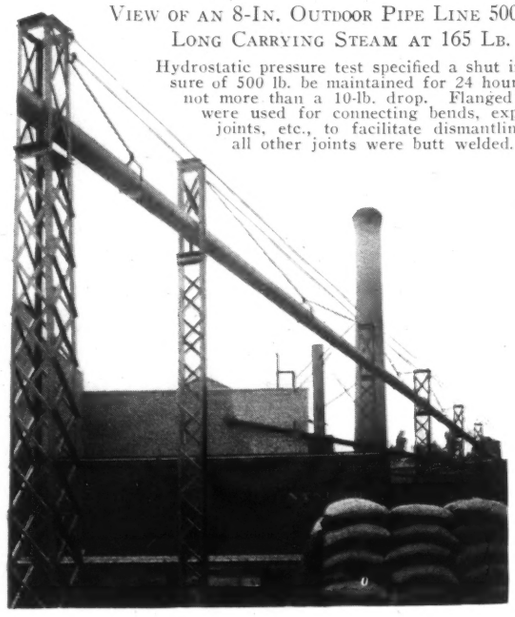
Several types of flanged joints and gasket facings were discussed in a previous article[2] and the 400-lb.-1,000-F piping installation at the Delray station of The Detroit Edison Company,[3] with its special types of flanged and welded joints and other unique design features, has also been described in considerable detail in this magazine.

Welded joints of the bolted-flange, seal-welded type have been used extensively on high pressure steam lines. Load-carrying welded joints, both with and without flange and bolt reinforcement, have also been used on high-pressure, high-temperature pipe lines. The progress that has been made in welding technique during recent years is establishing the dependability of this type of joint and its general acceptance in the comparatively near future under rather rigid inspection requirements appears to be indicated.

American Standard Joints

Because most of the high-pressure joints used for steam lines are within the pressure range of the *A. S. A.* standards and are either regular standard flanged joints or at least follow their general dimensional design with slight modifications, this discussion will be based on the American standard types of flanges. A first step in assembling a joint that will give satisfactory service on a high pressure steam line is to make sure that the flanges, bolts, etc., conform to the requirements of the standard. It consequently will be of interest to discuss briefly some of these requirements.

Flanges: When flanges are cast or forged integral with the wall of a fitting or a valve, they are stiffened and strengthened thereby, and the new American standard for loose flanges requires the use of substantial hubs



VIEW OF AN 8-IN. OUTDOOR PIPE LINE 5000 FT. LONG CARRYING STEAM AT 165 LB.

Hydrostatic pressure test specified a shut in pressure of 500 lb. be maintained for 24 hours with not more than a 10-lb. drop. Flanged joints were used for connecting bends, expansion joints, etc., to facilitate dismantling, but all other joints were butt welded.

[2] "Higher Pressures and Temperatures as They Affect the Design, Installation and Maintenance of Piping," HEATING, PIPING AND AIR CONDITIONING, February, 1931.

[3] "Piping for 1000-Degree Steam," HEATING, PIPING AND AIR CONDITIONING, April, 1931.

"Designing High Temperature Steam Piping," HEATING, PIPING AND AIR CONDITIONING, October and November, 1931.

which increase the stiffness and strength of the flange with comparatively little addition to its weight and with no addition to the length of the bolts.

American Standard B16e specifies that the flange material shall be in accordance with *A. S. T. M. Specification A-105.* This specification recognizes three grades of flanges designated as *Classes A, B,* and *C. Class A* flanges are made of low-carbon steel of suitable composition for forge welding to the pipe. *Class B* flanges are made of steel having a minimum ultimate tensile strength of 60,000 lb. per sq. in., and a minimum yield point of 30,000 lb. per sq. in. *Class C* flanges are made of steel having a minimum ultimate tensile strength of 70,000 lb. per sq. in., and a minimum yield point of 36,000 lb. per sq. in.

The *A. S. A.* standards are really based on the use of *Class C* flange material, but *Classes A* and *B* material are permitted if the flanges are provided with adequate hubs or are otherwise reinforced to compensate for their lower strength. Also flanges forged integral with or forge welded to the wall of the pipe are considered as being additionally reinforced. *Class B* flanges are extensively used and usually have been satisfactory in high-pressure steam service.

Bolts: The bolts specified for *A. S. A.* flanges are required to be heat-treated, alloy-steel bolt studs with a nut at each end. Headed bolts are prohibited because the drastic forging operation involved in the heading of the bolt has resulted in failures in a number of instances due to weakness at the junction of the bolt head and shank. The material from which the bolt studs are made must conform to *A. S. T. M. Specification A-96,* which recognizes three grades of steel designated as *Classes A, B,* and *C.*

Class A bolting material must have a minimum ultimate tensile strength of 95,000 lb. per sq. in. and a minimum yield point of 70,000 lb. per sq. in. *Class B* bolting material must have a minimum ultimate tensile strength of 105,000 lb. per sq. in., and a minimum yield point of 80,000 lb. per sq. in. *Class C* bolting material must have a minimum ultimate tensile strength of 125,000 lb. per sq. in., and a minimum yield point of 105,000 lb. per sq. in.

Class A bolting material has been used to some extent for working steam pressures up to 400 lb. per sq. in., *Class B* up to 600 lb. per sq. in. and *Class C* for higher pressures, but the present tendency seems to be to specify *Class C* as the minimum requirement for high-temperature service regardless of the working pressure.

Threads: The bolt threads are *American Standard Screw Threads, Coarse Thread Series, Medium Fit (Class 3),* except that eight threads per inch are specified for all bolt diameters 1⅛ in. and larger. This is done to facilitate the tightening of the larger bolts so as to secure the desired initial gasket pressures.

Nuts: The nuts specified are cold-punched or cold-pressed semi-finished carbon-steel nuts of American standard rough dimensions, chamfered and trimmed. While the American standard rough nut dimensions are specified, it has been the customary practice to use the U. S. standard nut dimensions in this class of service, and after considerable discussion it now appears likely that the U. S. standard nut dimensions will be added to the list of American standard nut dimensions for use on flange bolts. The standard also specifies that washers when used under nuts shall be of forged or rolled carbon steel. The special beveled washers that are sometimes used require more care in assembly to see that they are mounted right side up, and if the outside diameter is oversize it may interfere with the wall of the spot facing on the flange.

Finish of Flange Faces

The American steel flange standard does not specify the type of finish to use for flange facing, but this is important because it vitally affects the sealing of the joint between gasket and flange and should be of such a nature as to prevent slipping or blowing out of the gasket.

For asbestos gaskets not less than 1/32 in. thick and for pressures up to 600 lb. per sq. in. the "phonograph" finish has been extensively used. This finish as usually made consists of grooves 1/64 in. deep, spaced 1/32 in. apart. For high pressures and thin asbestos gaskets the "serrated" finish is suitable. The serrated finish is produced by the use of a medium tool finish with the surface further roughened by grooves 1/100 in. deep, spaced 1/16 in. apart.

Bolting and Gaskets

Bolts and Threading . . . . Computing Stress in Bolts . . . . Chart Shows Turning Moment Required to Produce Desired Bolt Stress . . . . Unit Gasket Pressure

Bolts for Flanged Joints

In all flanged joints the bolts for holding the joint together must be sufficiently strong and rigid to withstand the internal pressure in the pipe line acting against a closed valve, second, to resist the additional tensile stress introduced by bending moments due to pipe line expansion, and third, to maintain the requisite unit gasket pressure to prevent leakage. Also, in view of the fact that a comparatively small elongation of the bolts will relieve the unit pressure on the gasket and cause leakage the bolt stresses caused by internal pressure and pipe line expansion should be kept at relatively low values. The writer also believes that the bolts should be as short as possible because long bolts increase the total elastic elongation under load as well as the plastic elongation due to creep at high temperature if the safe creep stress is exceeded.

For working pressures up to 600 lb. per sq. in. and working temperatures up to 750 F, studs threaded the full length have been used extensively in steam engineer-

TABLE 1—AMERICAN STANDARD BOLT STRESSES-FORMULA NO. 1

Bolt Stresses by Formula: $\frac{\text{Area to outside of face}}{\text{Area of bolts}} \times \text{Pressure} = \text{Stress}.$

Pipe Size	Dia. R.F.	Area	400			600			900			1350		
			Bolts	Area	Stress	Bolts	Area	Stress	Bolts	Area	Stress	Bolts	Area	Stress
½	1⅜	1.485	.....	.....	.....	4–½	0.502	1775	.....	.....	.....	.....	.....	.....
¾	1⅝	2.237	.....	.....	.....	4–⅝	0.808	1661	.....	.....	.....	.....	.....	.....
1	2	3.142	.....	.....	.....	4–⅝	0.808	2338	.....	.....	.....	4–⅞	1.68	2530
1¼	2	4.909	.....	.....	.....	4–⅝	0.808	3658	.....	.....	.....	4–⅞	1.68	3950
1½	2⅜	6.492	.....	.....	.....	4–¾	1.21	3220	.....	.....	.....	4–1	2.20	3980
2	3⅝	10.32	.....	.....	.....	8–⅝	1.62	3830	.....	.....	.....	8–⅞	3.36	4160
2½	4⅛	13.36	.....	.....	.....	8–¾	2.42	3315	.....	.....	.....	8–1	4.40	4100
3	5	19.64	.....	.....	.....	8–¾	2.42	4875	8–⅞	3.36	5260	8–1⅛	5.82	4560
3½	5½	23.76	.....	.....	.....	8–⅞	3.36	4240	8–1	4.40	4850	8–1⅛	5.82	5510
4	6⅜	31.92	8–⅞	3.36	3790	8–⅞	3.36	5690	8–1⅛	5.82	4930	8–1¼	7.43	5790
4½	6¾	35.79	8–⅞	3.36	4260	8–1	4.40	4880	8–1⅛	5.82	5540	8–1⅜	9.23	5230
5	7⅝	42.72	8–⅞	3.36	5100	8–1	4.40	5830	8–1¼	7.43	5470	8–1½	11.22	5140
6	8½	56.75	12–⅞	5.04	4500	12–1	6.60	5260	12–1⅛	8.72	5850	12–1⅜	13.84	5530
7	9⅝	72.76	12–1	6.60	4420	12–1⅛	8.72	5010	12–1¼	11.14	5880	12–1½	16.84	5830
8	10⅝	88.66	12–1	6.60	5370	12–1⅛	8.72	6110	12–1⅜	13.84	5760	12–1⅝	20.15	5840
9	11⅝	106.1	12–1⅛	8.72	4860	16–1⅛	11.63	5450	16–1⅜	18.46	5180	.....	.....	.....
10	12¾	127.7	16–1⅛	11.63	4390	16–1¼	14.85	5150	16–1⅜	18.46	6220	12–1⅞	27.6	6250
12	15	179.7	16–1¼	14.85	4840	20–1¼	18.56	5820	20–1⅜	23.1	6990	16–1⅞	36.8	6580
14	16¼	207.4	20–1¼	18.56	4460	20–1⅜	23.1	5380	20–1½	28.1	6650	.....	.....	.....
15	17¼	233.7	20–1¼	18.56	5040	20–1⅜	23.1	6070	20–1⅝	33.6	6270	.....	.....	.....
16	18½	268.8	20–1⅜	23.1	4670	20–1½	28.1	5760	20–1⅝	33.6	7200	.....	.....	.....
17	19¾	306.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
18	21	363.1	24–1⅜	27.7	5240	20–1⅝	33.6	6480	20–1⅞	46.1	7090	.....	.....	.....
20	23	415.5	24–1½	33.7	4940	24–1⅝	40.3	6180	20–2	53.0	7050	.....	.....	.....
22	25¼	500.7	24–1⅝	40.3	5030	.....	.....	.....	.....	.....	.....	.....	.....	.....
24	27¼	583.2	24–1¾	47.5	4920	24–1⅞	55.3	6330	20–2½	85.8	6110	.....	.....	.....

ing practice and in some cases the central body portions of the studs have been turned down to the root diameter of the threads. The object of both these treatments is to eliminate the stress concentration occasioned by the change in cross-sectional area at the ends of the threaded sections, in order to reduce the danger of fatigue failure by having the reduced cross-sectional area extend uniformly over the full length of the stud.

However, with higher working pressures the bolts had to be made considerably longer in order to span the thicker flanges and the thicker joint structure between the flange faces, and a further tendency toward reduction of gasket pressure was introduced by the fact that higher working temperatures decrease the safe creep stress of the bolting material and increase the possibility of plastic elongation due to creep.

Consequently, when long bolts must necessarily be used at high temperature it appears to be a logical procedure to leave the central body portion of the stud unthreaded so as to limit the more highly-stressed material to as short a length as possible, and to provide against the possibility of fatigue failure by the use of high-strength bolting material. This is what was done at The Detroit Edison Company in their 400-lb.—1,000-F installation.

Computing Stress in Flange Bolts

Table 1 shows the usual method of computing the stresses in flange bolts, in which the bolts are assumed to be tightened just enough to resist the maximum steam working pressure and at the same time impose an equal unit pressure on the gasket. When computed by this method the total bolt load is taken as equal to the force exerted by the internal pressure acting upon the area circumscribed by the outside diameter of the gasket.

If the actual bolt stresses under operating conditions did not exceed the values shown in Table 1, the alloy steel bolts would be but slightly affected by creep at the temperatures to which they are subjected, but in actual practice the bolt stresses are necessarily higher because of the need for a higher initial gasket pressure in order to provide a margin of safety against leakage when the bolts elongate and the flanges deflect under operating conditions.

Turning Moment to Produce Bolt Stress

The turning moment on the nut required to produce the desired initial bolt stress is dependent upon a number of variable factors and consequently in making up a joint it is customary practice to tighten the bolts excessively in order to make sure that the joint will be

TABLE 2—INITIAL GASKET PRESSURES AND GASKET PRESSURE TO WORKING STEAM PRESSURE RATIOS FOR AMERICAN STD. FLANGES

Pipe Size In.	400			600			900			1350		
	Gasket Pres.		Ratio	Gasket Pres.		Ratio	Gasket Pres.		Ratio	Gasket Pres.		Ratio
	R.F.	T.& G.		R.F.	T.& G.		R.F.	T.& G.		R.F.	T.& G.	
½	.....	.....	.....	23600	.....	39.2	.....	.....	.....	.....	.....	.....
¾	.....	.....	.....	21600	.....	36.0	.....	.....	.....	.....	.....	.....
1	.....	.....	.....	16500	.....	27.5	.....	.....	.....	22700	.....	16.8
1¼	.....	.....	.....	7800	.....	13.0	.....	.....	.....	26700	.....	19.8
1½	.....	.....	.....	10000	.....	16.7	.....	.....	.....	22500	.....	16.6
2	.....	.....	.....	10800	.....	18.0	.....	.....	.....	31000	.....	22.2
2½	.....	.....	.....	11400	.....	19.1	.....	.....	.....	30000	.....	22.1
3	.....	.....	.....	7700	.....	12.8	.....	21200	23.6	28600	.....	21.0
4	6600	.....	16.5	6600	.....	11.0	.....	17400	19.4	20000	.....	14.8
5	5180	.....	12.9	5900	.....	9.9	.....	16500	18.4	20800	.....	15.4
6	6060	.....	15.2	6950	.....	11.6	.....	18600	20.8	24000	.....	17.8
8	5160	.....	12.9	5800	.....	9.7	.....	15400	17.2	19000	.....	14.0
10	6330	.....	15.8	6900	.....	11.5	.....	14200	15.8	15600	.....	11.6
12	5600	.....	14.0	6500	.....	10.8	.....	15000	16.8	17400	.....	12.8
14	6150	.....	15.4	6500	.....	10.8	.....	15400	17.2	.....	.....	.....
16	5450	.....	13.6	.....	11600	19.3	.....	12800	14.3	.....	.....	.....
18	5060	.....	13.9	.....	11300	18.8	.....	13300	14.8	.....	.....	.....
20	4960	.....	12.4	.....	10800	18.0	.....	11500	12.8	.....	.....	.....
24	4470	.....	11.2	.....	10700	17.9	.....	12500	13.9	.....	.....	.....

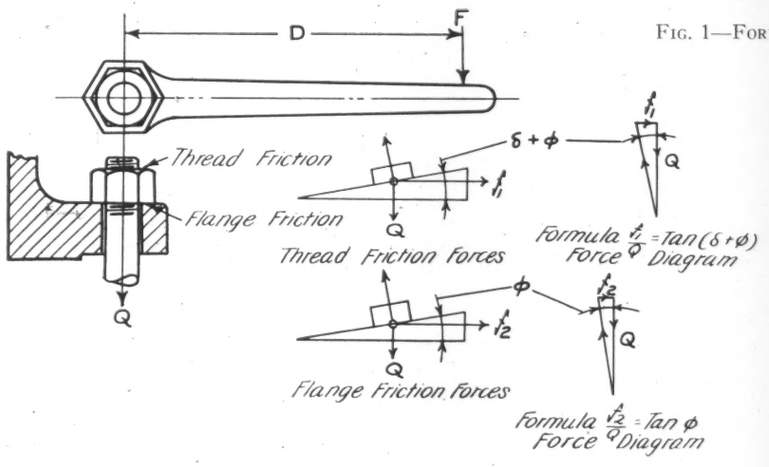


FIG. 1—FORMULA FOR COMPUTING TURNING MOMENT

Q = Total bolt load in lb.
f_1 = Rotating force at pitch of thread to overcome thread friction
f_2 = Rotating force at pitch of thread to overcome flange friction
f = Total rotating force at pitch line of thread
F_1 = Rotating force at end of wrench handle to overcome thread friction
F_2 = Rotating force at end of wrench handle to overcome flange friction
F = Total rotating force at end of wrench handle
D = Length of wrench handle, inches
P = Pitch of thread, inches
S = Pitch radius of thread, inches
δ = Angle of thread, degrees

$$\text{Tan } \delta = \frac{P}{6.28\,S}$$

ϕ = Friction angle, 10¾ degrees
Tan ϕ = 0.190 = coefficient of friction
FD = Turning moment, in.-lb.
Formula: $FD = (F_1 + F_2)\ D = (f_1 + f_2)\ S = QS\ [\tan\ (\delta + \phi) + \tan\ \phi]$

$$F = \frac{QS}{D}\,[\tan\ (\delta + \phi) + \tan\ \phi]$$

tight in service. Fig. 1 shows a method of computing the turning moments required to produce various initial stresses in bolts of different sizes and Fig. 2 shows a logarithmic chart of the turning moments required for bolts from ½ in. to 2½ in. inclusive and stresses from 10,000 to 60,000 lb. per sq. in. as computed by this method. The amount of turning moment required to produce the desired bolt stress is subject to considerable variation due to differences in the quality of finishes and fits of the friction surfaces and the effectiveness of lubrication, but Fig. 2 represents average conditions with the friction surfaces well lubricated.

While the chart is based on a theoretical formula a sufficient number of points on the curves have been checked against actual experimental data to give assurance that the chart closely approximates average field conditions.

A brief study of Fig. 2 reveals that in order to produce the same initial stress in all sizes of bolts the required turning moment increases rapidly with increase in bolt diameter. On the other hand the standard length of handle for commercial wrenches is approximately proportional to the bolt diameter, and in order to develop the desired initial stress in the larger bolts it consequently is necessary to increase the available turning moment by some method. This is usually accomplished by moderately "sledging up" the nuts and even then the initial stress in the larger bolts is considerably less than it is in the smaller bolts.

The line AA and tabulation in Fig. 2 indicate the normal initial stresses for the various sizes of bolts as computed by the formula[4]

$$t = 30{,}000 \div d$$

in which t equals the unit tensile bolt stress in lb. per sq. in. and d equals the diameter of bolt in inches.

The high initial bolt stresses shown in Fig. 2 would be cause for apprehension of failure due to creep at high temperature were it not for the fact that a slight amount of creep in the bolts and flanges relieves a large portion of the excess gasket pressure and reduces the stresses to safe limits before the gasket pressure is lowered enough to permit leakage. Table 2 shows the initial gasket pressures and ratios of gasket pressure to working steam pressure for the *American Standards Association* flanges series, 400 to 1,350 inclusive, based on the initial bolt stresses shown in Fig. 2. Assuming that the initial gasket pressure should be at least ten times the working pres-

[4] See "Piping Handbook," p. 345.

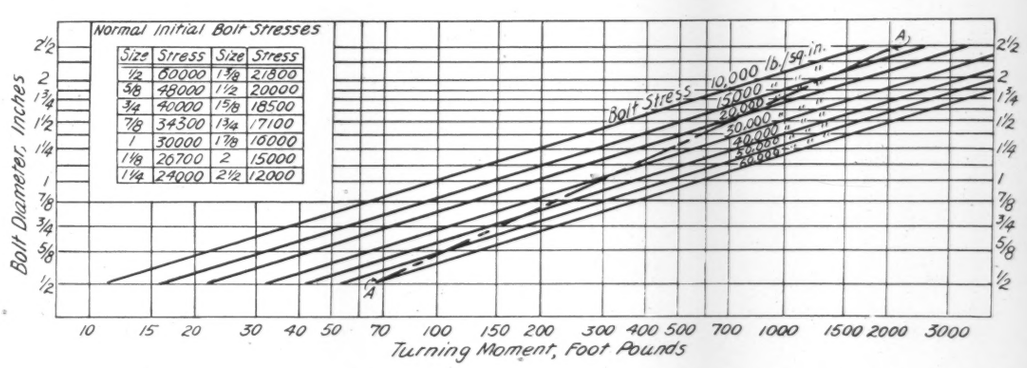


FIG. 2—CHART SHOWING INITIAL BOLT STRESSES AND RELATION BETWEEN TURNING MOMENTS AND BOLT STRESSES

sure, Table 2 indicates that for series 600 sizes 16 in. and larger, series 900 all sizes, and series 1,350 sizes 1¼ in. and larger if the raised face is used the width of the gasket should be reduced to secure a higher unit pressure. When a narrow gasket is used on the raised face joint it is customary to decrease the area of the gasket by increasing its inside diameter so that it will contact with the flange on the outer portion of the raised face. The outside diameter of the gasket is usually brought out to the bolts in order to aid in centering the gasket when assembling the joint.

It will also be noted in Table 2 that the unit gasket pressures on the different joints range from less than 5,000 to more than 30,000 lb. per sq. in., which indicates that the type of gasket used needs special consideration. The primary function of a gasket is to provide an impervious material relatively softer than the flanges themselves between the contact faces of the flanges so that the gasket material will be compressed when the flange bolts are tightened, filling up the small irregularities of the space between the flange contact faces and permanently sealing the joint.

It is obvious that the unit gasket pressure required to compress the gasket and seal the joint varies considerably for different types of gaskets. If the unit pressure is too high the gasket flows out from between the contact surfaces and leakage occurs, but in cases of this kind it has been found that reducing the thickness of the gasket will correct this difficulty by permitting the flange faces to actually come in contact at the points of extreme gasket pressure, thus providing a positive bearing and preventing any serious localized reduction of gasket pressure at other portions of the contact surface due to pipe line strains.

It therefore is evident that in addition to the design of the joints, careful consideration must also be given to the gasket material, width and thickness. Compressed asbestos gaskets as thin as 1/100 in. have been successfully used in this type of service, and, while gaskets of this kind must be handled with care, they can be subjected to very high unit pressures. With very thin gaskets rigidity of flanges is essential in order to prevent localized reduction of gasket pressure between bolts, and the flange faces must be accurately machined.

Assembly and Testing of Joints

Order of Tightening Bolts for Uniform Gasket Pressure Shown by Diagram . . . . Methods of Field Testing Joints . . . . Ratio of Test Pressure and Working Pressure . . . . Test Procedure

Assembling of Joints

In making up the joints all contact surfaces should be perfectly clean and the gaskets should be accurately positioned so they will bear uniformly over the entire gasket surface. The bolts may be tightened with a long-handled ratchet wrench and for ⅞ in. and larger sizes it is usually necessary to tighten them further by the use of a stub wrench and a maul in order to secure the desired initial gasket pressure. The crew assembling the pipe line usually puts in only enough bolts to hold the flanges securely in place and the installation of the rest of the bolts and the final tightening is done by a special crew.

The order of tightening the bolts is important because when they are simply tightened in succession around the circle the gasket pressure will not be uniform. It would be ideal from the standpoint of uniform gasket pressure if all the bolts could be tightened simultaneously, but as this cannot be done the next best thing is to approach the ideal as closely as possible. A good procedure is to tighten all the bolts by hand, then tighten two bolts 180 degrees apart, then advance clockwise 90 degrees and repeat the operation, continuing in this way until all the bolts are tight.

Fig. 3 illustrates the order of tightening the bolts in a 16-bolt flange by this method. It will be noted that the tightening of the first four bolts, which are located 90 degrees apart, establishes the general position of the flange and makes possible general uniformity of gasket pressure. The subsequent tightening of the other bolts affects the gasket pressure only in localized areas.

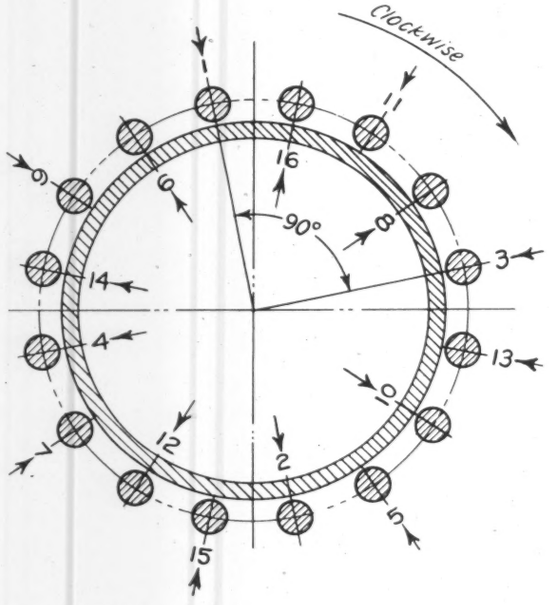
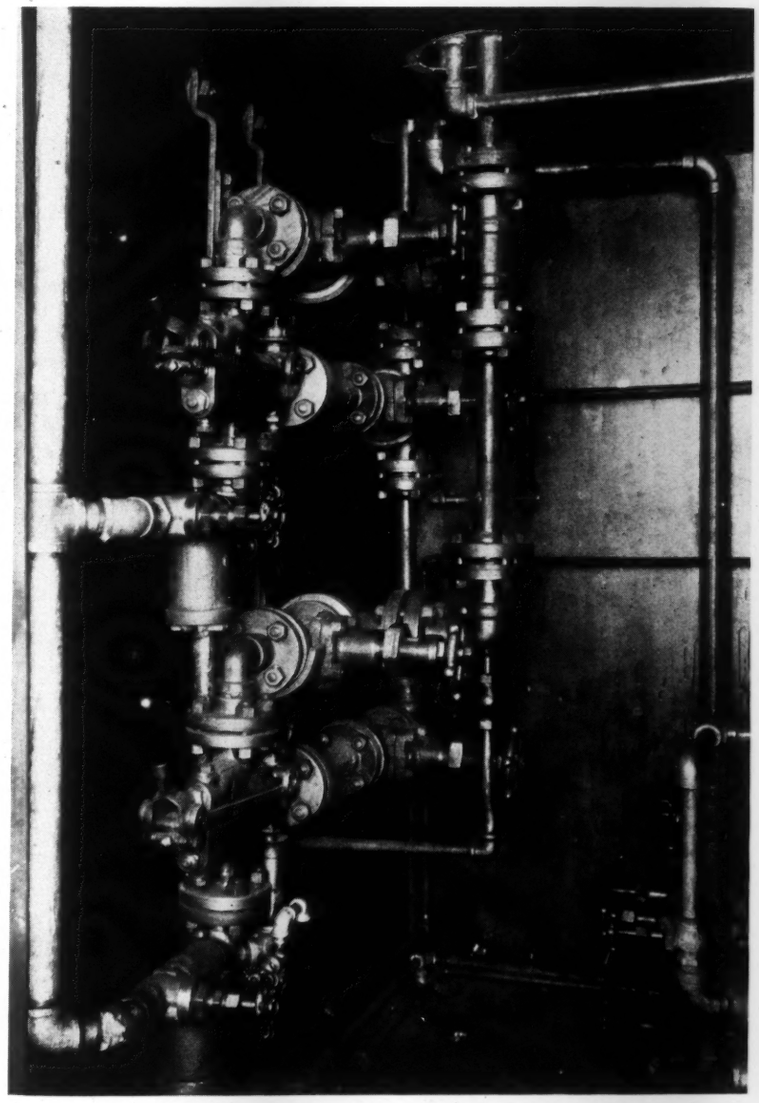


Fig. 3—Order for Tightening Bolts for 16-Bolt Flange

When two bolts diametrically opposite each other are tightened it is obvious that the portion of the gasket adjacent to the first bolt tightened, which may be called the entering bolt, will be subjected to a somewhat higher pressure than the portion adjacent to the second bolt, which may be called the leaving bolt. However with the order of tightening as illustrated in Fig. 3 the entering and leaving bolts alternate around the circle except on one diameter, and this distributes the gasket pressure as uniformly as is possible in actual installation



Injector Piping for a 900-Lb. Boiler. Note the Screwed Joint Low-Pressure Piping on the Suction Line. High-Pressure Flanged Joints Are of the Asbestos-Gasket Type. Screwed Joints, Back Welded, Are Also Used on the High-Pressure Piping

work. While this routine is somewhat more intricate than simply tightening the bolts in sequence it can be carried out readily and a more uniform gasket pressure is secured.

In this connection it is probably desirable to introduce a word of caution against excessive tightening of bolts. When the flanges and bolts are fully up to the requirements of the American standard, the type and finish of flange faces, and the type, width and thickness of gaskets are properly chosen, dishing of flanges and over-stressing of flange bolts should not be necessary in order to secure tight joints. The procedure followed in the tightening of bolts should be governed accordingly.

So far as the writer knows the only method of accurately determining the stresses set up in the bolts is actually to measure the elongation of the bolts. The modulus of elasticity of the alloy-steel bolting material at atmospheric temperature may be taken as 30,000,000 and using this value the elongation of the bolts may be converted directly into stress if the elastic limit of the material is not exceeded.

Field Testing of Joints

It is necessary to subject high-pressure steam lines to an internal pressure test before placing them in service in order to make sure that all joints and other parts of the system are tight and capable of withstanding the working pressures to which they will be subjected.

All valves are thoroughly tested at the factory before shipment, but the test after being mounted in the pipe line is also desirable so as to determine definitely that they have not been injured in any way subsequent to the factory test.

The increase in steam working pressures has been so rapid during recent years that the development of standard, universally accepted pipe line erection and testing methods quite naturally has lagged behind because of lack of a sufficient background of experience in this new field.

The usual method of testing is to subject the system to a hydrostatic pressure of from 1½ to 2½ times the steam working pressure and a steam test at the full working pressure also may be required. For moderately high pressure systems of around 200 to 300 lb. the hydrostatic test pressure has often been 2½ times the working pressure, but for the higher pressure it is seldom considered desirable to subject the system to a test pressure in excess of twice the working steam pressure.

The present trend of engineering thought along these lines is to make a distinction between indoor and outdoor piping and to specify a more exacting test for the indoor piping because of the greater hazard to human life and the possibility of large property damage and extended interruption of service if a failure should occur.

Test Pressure Must Be in Proportion to Working Pressure

The test pressures which are favored at present for these two piping divisions are twice the working steam pressure for indoor piping and 1½ times the working steam pressure for outdoor piping. While higher hydrostatic pressures would be somewhat more effective in detecting leaks, they increase the bolt and flange stresses and decrease the gasket pressures so much that a joint designed to hold the test pressure may be badly out of proportion for the working pressure at which the pipe line is intended to operate.

To illustrate this, consider the case of an *A. S. A.* 12-in., 600-lb. joint. The initial stress in the 1¼-in. bolts as shown in Fig. 2 is 24,000 lb. per sq. in. The corre-

sponding initial gasket pressure is 6,500 lb. per sq. in. and the ratio of the initial gasket pressure to the working pressure is 10.8. (See Table 2.) If a test pressure of, say, 3½ times the working pressure (2,100 lb. per sq. in.) were specified the gasket pressure would be reduced to 3,180 lb. per sq. in. and the ratio of gasket pressure to internal pressure would be reduced to 1.5. This ratio of gasket pressure to internal pressure is quite evidently too low and in order to insure against blowing out of the gasket it would be necessary to increase greatly the initial gasket pressure by excessively tightening the bolts.

Because of the higher initial gasket pressure, this would doubtless require the use of a special gasket suitable primarily for the high test-pressure rather than for the working pressure at which the joint is intended to operate. For this reason, and also to reduce the danger of over-stressing the bolts and flanges, it is considered good engineering practice to limit the test pressure to not more than twice the working pressure.

Hydrostatic Test Procedure

After the pipe line is erected and the joints are properly made up, the next step is the application of the hydrostatic pressure test. In order to facilitate the locating of any leaks that may develop it is desirable in the case of extensive piping systems to subdivide them into convenient sections and test each section separately. The section under test is filled from the service pressure mains, great care being taken to see that all the air is vented out and the entire internal volume is completely filled with water.

The pressure is then raised to the specified test pressure by means of a hand pump, after which one or more valves in the line between the pump and the test section of piping are tightly closed. Special attention must be given of course to valve stuffing boxes, the dead ends of the section, branch connections, outlets and all other points where leakage might occur, because the test specifications are based upon absolutely no leakage and usually require that the test pressure shall be maintained within two per cent of its original value for 24 hours.

In view of the fact that the coefficient of compressibility of water is so small as to be entirely negligible for the small pressure drop permitted in the specification, it is readily apparent that this permissible pressure drop is really intended to cover pressure variations due to temperature changes rather than leakage.

Drying Out of Valve Packing

Valve stuffing box packings are sometimes troublesome in tests of this character. This may be due to drying out and air-hardening of the packing during the period the valves have been in stock subsequent to the factory test. However the type of packing required for high temperature service is rather hard at room temperature and even when the packing is new it is sometimes difficult to make it hold absolutely bone dry on a high-pressure cold water test. In order to overcome this difficulty it is becoming rather common practice to use one or two rings of softer packing near the top of the stuffing box, with several rings of the harder packing below and at least one ring above them. High temperature steam valves are designed with condensing chambers and rather deep stuffing boxes so that the softer packing located as described is not subjected to the extreme temperature and it serves a useful purpose in preventing leakage when the line is cold.

In blanking off the outlets, branch connections and dead ends, it is desirable to have some means of determining that no leakage is occurring at these points, because when difficulty is encountered in holding the pressure it sometimes involves a long and tedious search to discover the source of the leakage.

No general rules can be laid down for governing these points because of the great variation in conditions that may arise, but it behooves the erection superintendent to make absolutely sure that no leakage is occurring through hidden connections.

Testing Will Become Standardized

While the methods of testing joints of high pressure steam lines have not been definitely standardized as yet, much progress along these lines has been made, and as the working pressures and temperatures in common use become stabilized and more experience is gained in testing procedure, definite standardization will no doubt be accomplished.

Note: The writer wishes to express his appreciation to Mr. R. H. Baker for the piping illustrations and for helpful suggestions in the preparation of this article.

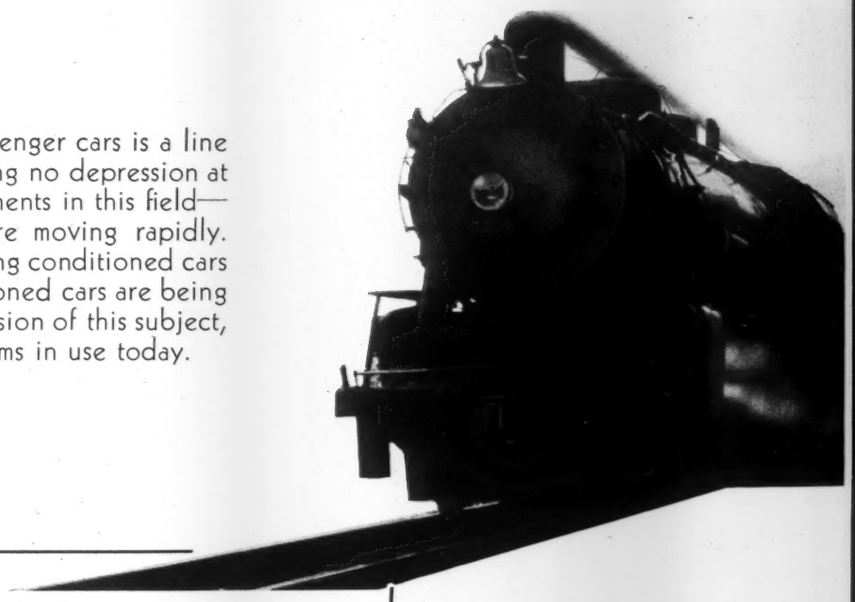
Effect of Temperature on Metals Subject of Symposium

The "Symposium on Effect of Temperature on Metals," published jointly by the *American Society for Testing Materials* and the *American Society of Mechanical Engineers,* contains the papers (including the discussions) which were presented at the joint meeting of the two societies held in Chicago, June 23, 1931. According to H. W. Gillett, in his introduction to the volume, as a result of the first symposium on this subject in 1924, the joint research committee on effect of temperature on properties of metals was established; this committee felt that because of the progress made in the last seven years the time was ripe for a summation of the present pressing demands of the engineer, and of what the metallurgist has produced to meet these demands.

The work is divided into two general sections, the first on engineering trends and requirements for metals at high and low temperatures and the second on properties of available metals for high- and low-temperature service. Among the topics covered in the former are requirements of the power plant industry, steam turbine materials, requirements for high-temperature piping materials, trend of progress in Great Britain, mechanical properties of metals at elevated temperatures, and the requirements of various industries. The second section contains sixteen papers on the properties of various steels and alloys.

The book consists of over 800 pages and is carefully indexed. Its size is 6x9 and it is attractively and substantially bound. It may be obtained from the *A. S. T. M.* at 1315 Spruce st., Philadelphia, or the *A. S. M. E.*, 29 W. 39th st., New York City.

The air conditioning of railway passenger cars is a line of activity which seems to be suffering no depression at the present time; in fact, developments in this field—both technical and commercial—are moving rapidly. Several railroads are already operating conditioned cars and trains, and several more conditioned cars are being built. This article is a general discussion of this subject, and it describes some of the systems in use today.



Car Conditioning Developing Rapidly

AIR conditioning and cooling of railway passenger cars is a development which has commanded the increasing attention of heating and air conditioning engineers and of the public in the last five—and particularly in the last two—years. Although it is relatively new, passengers having experienced summer comfort on railway cars patronize in large numbers the roads on which it is provided, and differentiate between the kind of summer atmospheric comfort on different roads.

With the United States having 32 per cent of the railway mileage of the world,

with 52,100 passenger cars in the United States which travel approximately 700,000,000 passenger-miles per year,

with approximately 60,000 persons occupying Pullman berths each night,

with a decrease of 24 per cent in passenger traffic since 1924,

with increasing demands for better atmospheric conditions in railway passenger cars both summer and winter,

with the statement of leading roads that there is nothing within reason that they will not do for the atmospheric comfort and health of passengers,

with the belief on the part of prominent railroad officials that air conditioning will bring back a large part of the railway travel lost to other modes of transportation,

air conditioning and cooling for railway passenger cars promises to be an engineering development of increasing importance. It is stated that ninety-seven railroads are now considering air conditioning and it is predicted that the summer of 1932 will see a number of additional trains so equipped.

This article has mainly to do with air conditioning and cooling. But let us first trace hastily the history of railway passenger car heating and ventilating in this country. With this background we can better appreciate and understand the present achievements in cooling and air conditioning.

Methods of Passenger Car Heating

When railway passenger cars were first heated crude cast-iron, wood-burning stoves were used. Then came the coal-burning stove followed by a coal stove encircled by a hot air chamber. About 1870 a closed hot-water system made its appearance, steam jackets being added later. Shortly after this what was known as "pressure" or "straight steam" heating system was introduced in car heating. This was developed into the vapor system about 1903, and while many improvements have been made, the same fundamental principles have been used for many years in heating cars in this and many foreign countries.

In England during 1930 and 1931 four first-class coaches and five sleeping cars were equipped with what might be termed a warm-air system of heating and ventilation. Electric fans force air over steam coils and through ducts into the corridor. Air in the car may be recirculated or new air may be taken from outside. Branch ducts lead to the various compartments. The rate of discharge is under control of the passengers who can direct the flow of air in any desired direction.

A Car Precooler Used by the B. & O. A Fan Draws Air from the Outside Across Ice Cakes in the Truck, Discharging It into the Car Through a Window. The Warm Air in the Car Passes Out Through Over-head Ventilators

The motor operating the fan requires 0.15 hp., current being supplied from the 24-volt lighting circuit.

Heating Cars in Yards

Passenger, express and mail cars awaiting occupancy are regularly preheated in cold weather by means of flexible pipe connections with the yard steam line, by steam from a locomotive, from a stationary boiler, from a steam accumulator, by placing the cars in heated buildings, and in some instances by a coal stove or a portable emergency stove.

The cost of heating cars standing at terminals is a substantial part of the total heating cost; in many cases a large part of it is unwarranted. It is not uncommon to find cars heated to 80 F or 100 F in extreme cases while lying in the yards. A temperature of 50 F is considered high enough for washing, cleaning and prevention of freezing of toilets and water pipes and damage to the finish. Much steam may be saved by suitable temperature regulators.

Average steam consumption of steel cars standing in the yards is 1-7/10 lb. of steam per hr., for each degree of difference between inside and outside temperature. Following is a result of a test in a 70-ft. steel passenger car with windows, doors and ventilators closed, all admission valves open, and standing in a yard temperature of 20 F.

Inside Temperature of Car	*Steam Condensed per Hour*
50 F	30 lb.
70 F	60 lb.
90 F	120 lb.

Coal consumption at 70 F inside and 20 F outside was approximately 300-350 lb. for 24 hours, no temperature controls being used. Under running conditions condensation was found to be about double this amount.

Heating Cars in Motion

For a steel car running 30 m.p.h., 3 to 3½ lb. of steam per hr. are usually required per degree difference between inside and outside temperature. Approximately twice the amount of radiating surface per cubic foot is used in railway passenger cars as is used in dwellings. Cars operating near the equator do not have heating equipment. Gas-electric cars are often heated with a hot-water system. Trains going into New York terminals pulled by electric locomotives are heated by a flash-steam boiler carried on the locomotive. This system is also used on gas-electric cars and for heating trailers. The flash-steam boilers are equipped with oil burners with automatic control. Practically all passenger, mail and express cars in Canada, the United States and Mexico, are heated with vapor steam or hot water.

Precooling Passenger Cars

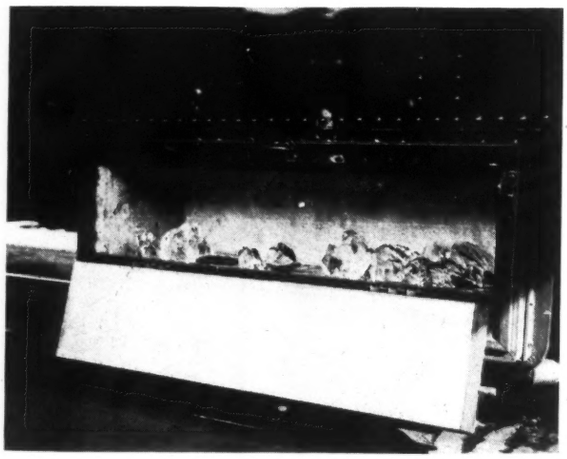
Recently increasing interest has been manifested in *precooling* of passenger cars. Tests have shown that even in temperate latitudes, the temperature of the air in cars standing in yards under the summer sun will reach 100 F or more. Passenger cars have been and are now precooled by placing cars in tunnels or under shelters away from the sun with doors and windows open, by means of portable blowers which blow heated air out of the cars, by portable blowers which force ice-cooled air through the cars, by forcing air over ice containers in the car, by blowers which force refrigerated air through cars, by spray systems in the cars and by operation of the refrigerating systems in the cars themselves.

Conditioning Cars with Ice

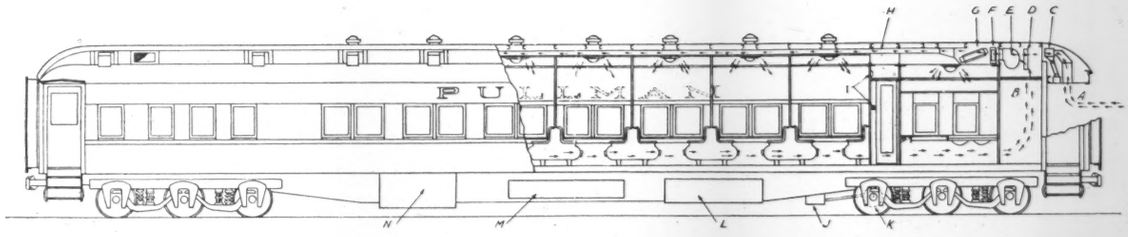
An ice-cooled dining car on the Pennsylvania Railroad is equipped with ice bunkers in which are placed seven three-hundred pound cakes of ice. The bunkers are flooded with water which passes through a strainer to a pump underneath the car and is pumped to the cooling coils between the roof and the ceiling at one end of the car. This is used between New York and Washington.

Equipment for cooling cars by means of ice while standing still as well as in transit consists of two insulated compartments for ice attached to the bottom of the car with openings outside for filling with ice and openings inside the car. A pump, two fans and four extended-surface radiators and air filters are also part of the equipment. The ice is sprayed with water which is pumped through the radiators. Moisture is condensed on the extended surface radiators, keeping them wet. Air is drawn through the radiators at 250-ft. velocity and is discharged at 1,000 ft. per min. at the end of the car. No eliminators are necessary.

Each compartment contains two radiators, a blower



Ice Compartment at the End of a Run, on a Car Which Is Conditioned by Means of Ice



PULLMAN METHOD OF CONDITIONING AIR IN RAILWAY CARS

A, fresh air drawn in from outside; *B*, air from inside of car drawn into duct for conditioning; *C*, dirt collector; *E*, fan; *F*, heating unit used in winter; *G*, cooling coils; *H*, distribution ducts; *I*, controls; *J*, generator which supplies power for operation; *K*, direct-drive for generator; *L*, refrigeration machine; *M*, brine tank from which cooled liquid is circulated to cooling coils; *N*, battery box for lighting system.

and an electric motor. In some instances a flat air filter is used. The equipment consists of:

1 450 ampere-hour battery (std. equip.)
1 4-kw. generator (std. equip.)
2 ice tanks, each of 800 lb. ice capacity
1 centrifugal pump—⅜ hp.
1 single hp. motor
2 felt filters
2 blowers, 1,000 c.f.m. ¼ in. with ¼-hp. motor each
4 extended surface copper radiators 24 x 18 in. by 3.24 in., 160 sq. ft. each

There are no automatic controls or dampers. Static pressure is maintained in the car at all times. In summer weather conditions, approximately 350 lb. of ice per hr. per car are used, equivalent to 5 tons refrigeration.

Early Air Conditioning of Cars

Some of the first experiments in air conditioning railway passenger cars were made in Chicago in 1924, and fostered by the Atchison, Topeka and Santa Fe Railway. However, complete air conditioning, that is, control of temperature, humidity, motion and cleanliness of the air was first applied to a 74-ton steel car No. 5275 on the B. & O. Railroad on July 1, 1929. In April, 1930, the B. & O. placed in service the first mechanically-operated air conditioned dining car, the *Martha Washington*. In May, 1931, the B. & O. ran the first completely air-conditioned train, the *Columbian*, from New York to Washington. Each car has its own air-conditioning unit so that any car can be taken from the train without affecting the others.

Since that time air conditioning has been applied to two trains on the B. & O. from New York City to Washington and to certain cars and trains on the B. & M., A. T. & S. F., M. K. & T., Pennsylvania and to Pullman cars. The B. & O. now has 38 air conditioned cars.

Some of the Factors to Be Considered

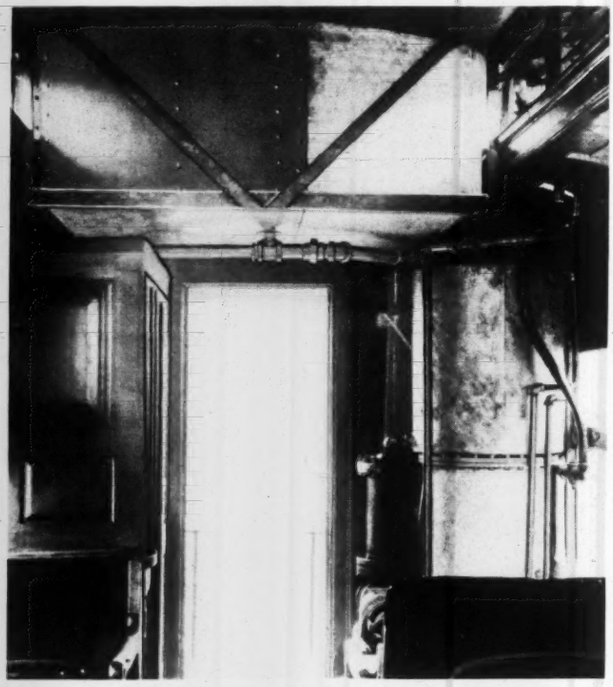
The air conditioning or cooling of railway cars requires careful study. Some roads running east and west can release all the water in their cooling and heating coils during cold or temperate weather, whereas, cars

INTERIOR OF THE PULLMAN MATANZAS WHICH IS EQUIPPED WITH AN AIR CONDITIONING SYSTEM. OUTLETS FOR CONDITIONED AIR MAY BE SEEN IN THE CEILING

running north and south are more likely to encounter decided changes in temperature. Passengers for Los Angeles may enjoy relatively cool weather in spring, summer or fall in the vicinity of Chicago, but encounter much higher temperatures by the time they reach Albuquerque and sometimes still higher temperatures in crossing the desert and upon approaching California. Passengers may leave a warm climate in the south and in a day or two enter a cold area in the northern part of the United States. Sometimes a cold or a warm wave covers a large part of the United States both north and south at one time. Railroads using air conditioning or summer cooling must be ready at all times for these temperature changes.

One problem of importance in air conditioning railway cars—and it is a problem in theater and other building air conditioning—is to so balance the conditions in order that excessive shock and discomfort will not be experienced by a passenger upon leaving the air conditioned car. This would seem particularly important where only one car in a train (the diner, for instance) is conditioned.

Heating of railway passenger cars is necessary during six or eight months each year in the northern parts of the United States and less than six months each year in the South, depending upon the latitude. On roads providing air conditioning or cooling in summer, the



ONE END OF A RAILWAY CAR, SHOWING THE DEMONSTRATION SET-UP OF A STEAM EJECTOR CONDITIONING SYSTEM. MAKE-UP WATER TANK AND EJECTOR ARE SHOWN

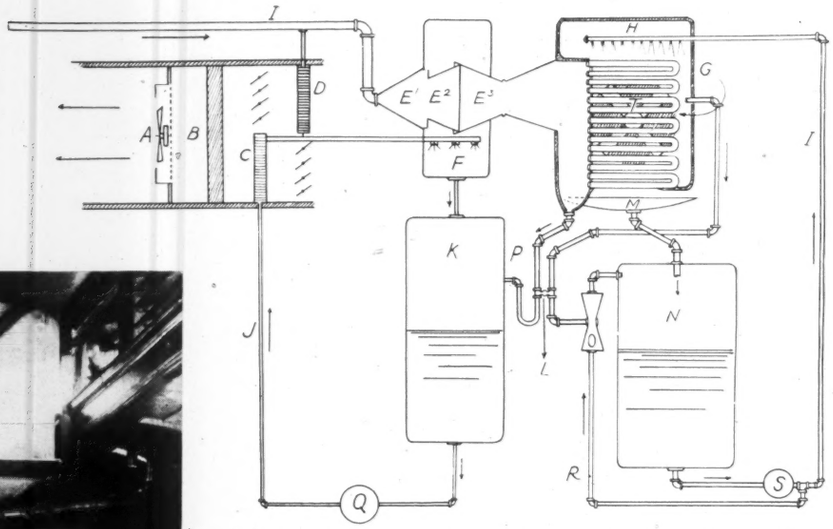


DIAGRAM ILLUSTRATING THE OPERATION OF A STEAM EJECTOR SYSTEM OF CONDITIONING

OPERATION OF THE SYSTEM: In operating the ejector system, the tank *K* is partly filled with water. The operator pushes an electric button and a lamp is lighted indicating that the system is ready for operation. The button starts circulating fan *A*. Dampers are adjusted, for when closed the system will not start. Then pump *S* for the condenser water and fan *T* for the cooling tower are started. At this point the operator turns on the valve permitting steam to pass through the strainer and separator in the steam line *I* and enter into the system through the ejector *E*. The ejector pulls a vacuum on tank *K*, cooling the water to 40 or 50 F, as required.

Steam from the locomotive being forced at high velocity through nozzle *E*, entrains water and vapor from *K* and *F*. Kinetic energy causes the steam and water vapor to be compressed into condenser coils *G*, which are a part of the cooling tower. Condensate water from make-up tank *N* is sprayed over the condenser coils and heat is removed by the evaporation of the condenser water from the surface of the condenser tubes, returning it to tanks *N* and *K*.

A second ejector *O* is so constructed that it takes all the condensate it can get, delivering it to make-up tank *N*, the remainder flowing into evaporator tank *K*.

The cold water pump *Q* draws cold water from evaporator tank *K* and circulates it through cooling coils *C* where it removes heat from the air which circulates through the car. This water, having absorbed heat from the air, is returned to evaporator tank at *F*. The spray being warm, it vaporizes rapidly under the vacuum influence of ejector *E*, and the cycle is complete.

The amount evaporated from the condenser tubes in the tower is nearly equal to the amount of steam and water vapor condensed. This nearly equal quantity is returned to make-up tank *N*, and to evaporator tank *K*. Make-up water is added as required, the amount depending upon the temperature of the outside air.

equipment is in operation approximately nine months each year.

Space is an important factor; it is usually limited so special attention is given to the size of equipment. Manufacturers strive to design equipment which is quiet, safe, light in weight, to use as little equipment as possible and make equipment which can utilize the power which is cheapest, most dependable and available whether the train is running or standing still, in train, or detached. Their object is to design equipment so simple in construction and operation that the average trainman will find no difficulty in its operation. Equipment must be designed to stand hard usage with minimum repair or maintenance costs.

An Air Conditioning System Used on a Number of Railroads

For air conditioning or cooling of railway passenger cars in transit, the following methods have been or are now employed: Forcing air over ice or pumping ice-cooled water through coils, the ice being held in containers in the car; by spray systems and by refrigerating systems, either compression or vacuum.

In one type of air conditioning system used on the B. & O. and other roads the cooling tower is located on one side of the vestibule entrance, the door being closed for this purpose. The remainder of the equipment is located in the half deck or suspended underneath from the supporting structure of the floor.

Among the types of power used are: (1) A compressor, brine and water circulating pumps are driven mechanically by a gasoline engine while the fans which serve the cooling tower and distribute cooled air are driven by motors in the electric lighting circuit which receives its power from the car axle. (2) The second type consists of a gasoline-engine generator set, the equipment being actuated by electric power from this source. (3) The third type is a body-hung axle-driven electric generator. In the first type a rotary pump is driven by a gasoline engine which circulates the brine while in the second and third types the pump is driven

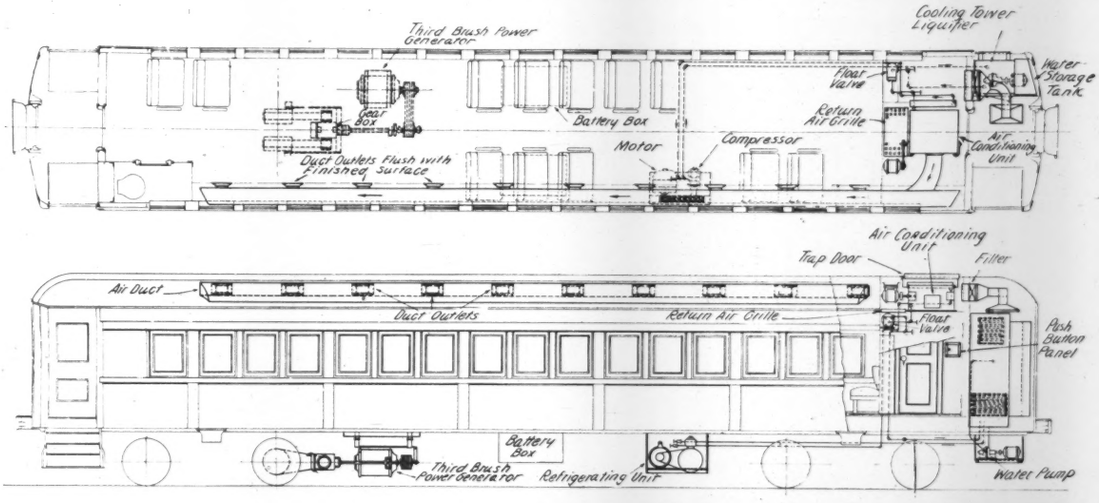
by a ¾-hp. motor. The first and second types of power units required a fuel supply tank which is protected by a tank of carbon tetrachloride connected to the tank in such a manner as to dilute the gasoline and make it noninflammable. Power has also been obtained by using small turbines supplied with steam from the locomotive.

Other equipment consists of a rotary compressor, cooler, condenser, pumps, cooling tower, cooling tower fan, cooling coils and ventilating fans. The fan for the cooling tower draws air through a water spray. The ventilating fans circulate air at the rate of 3,000 c.f.m. The fans which distribute air operate under three-speed manual control to regulate the temperature according to the weather. They distribute air through the opening in the upper deck panel of each side of the car, where the exhaust ventilators were formerly located. The cooled air passes to the floor in the center aisle at low velocity and then is circulated back to the fans through a return air grille located in the ceiling near the end of the car.

With the axle-generator system the power is cut in when the train is running 30 m.p.h. and is cut out automatically when the speed drops to 20 m.p.h. When the train slows down, the ventilating fans are operated from a storage battery and continue to circulate the cool air while the refrigerating apparatus is shut down. Flow of the refrigerant to the cylindrical brine cooler (6 ft. 5¾ in. long by 12¾ in. in diameter) is controlled by a float.

A Steam Ejector Air Conditioning System

Another system for air conditioning railway cars uses steam direct from the locomotive while running or standing still and steam from an outside source when cars are detached from a locomotive. This system (for the purpose of this article) may be termed the steam ejector system. It is designed to control the temperature, humidity, cleanliness and distribution of the air in summer and for the colder seasons to provide humidity, cleanliness and movement of the air and to supply a part of the heat, the remainder to be supplied by the regular floor heating coils or to be used as a protection against freezing of pipes when the air circulating unit is shut down. Warmed or cooled air from this system is delivered through ducts at the top and one end of the car.



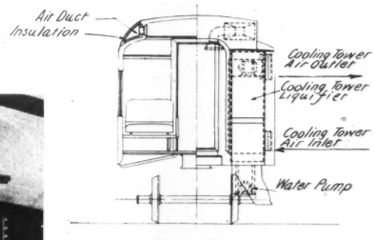
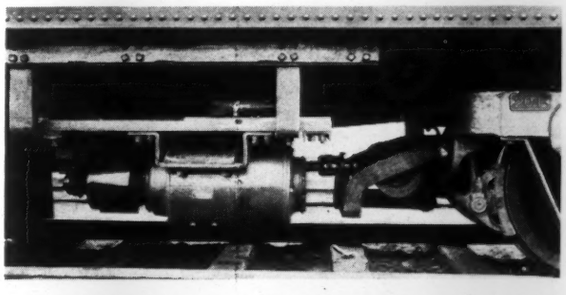
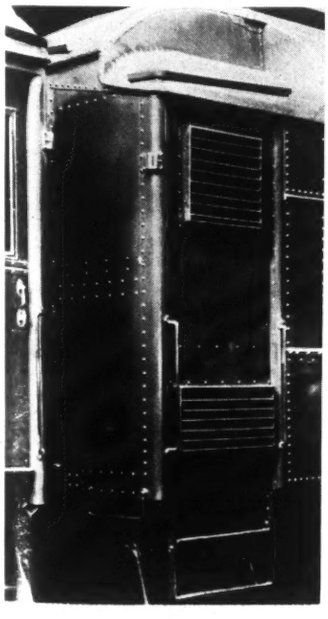


DIAGRAM SHOWING EQUIPMENT USED WITH ONE METHOD OF CONDITIONING



RAILROAD REFRIGERATING UNIT DESIGNED FOR OPERATION WITH F-12 AND MOTOR WITH HIGH- AND LOW-PRESSURE CUT-OUTS, MOUNTED BENEATH THE CAR

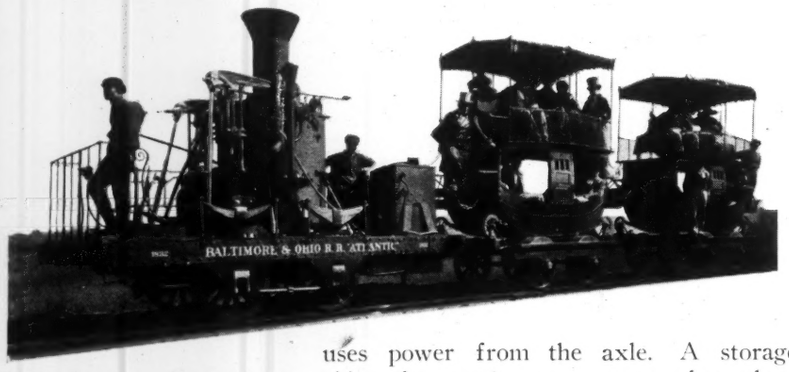


EXTERIOR OF COOLING TOWER LIQUEFIER OR CONDENSER

Principle of the Steam Ejector Method: Operation is based on the principle which causes water to boil or give off its heat and so become cooler as the atmospheric pressure on it is reduced. In other words, the temperature of the water corresponds to the pressure on it. When the absolute pressure over water is reduced to

0.178 lb. per sq. in., the temperature of the water is 50 F, or when the absolute pressure is reduced to 0.089 lb. per sq. in., the temperature of the water is 32 F. The lowering of the pressure is produced by the passage of steam at any temperature through the ejector.

Steam Ejector Location, Dimensions and Capacity: The ejector system equipment is concealed in the roof at the ends of the car. It supplies 2,000 c.f.m. of conditioned air affording a complete change of air each minute. The refrigerating capacity is 5 tons of melting ice for each 24-hour period. It is designed to remove 1,000 Btu per min. or 60,000 Btu per hr., which is equal to the heat normally dissipated by 150 persons. The complete air conditioning unit weighs about 5,000 lb. Steam consumption of the ejector is about 3 lb. per min. or about 170 lb. per hr., under maximum conditions.



Although the Passengers on a Train Such as This One Undoubtedly Got Enough "Fresh" Air, It Is Unlikely That Their "Atmospheric Comfort" Was Equal to That of the Passengers on a Modern Train Supplied with Clean, Cool and Properly Circulated Air

Line losses may add another 25 lb. per hr. Less steam is required for cooling in summer than for heating in winter. Water consumption is 12 to 14 gal. per hr. for a hot day. Twenty-five to thirty-five per cent outside air is used. Each of the two main units installed measures approximately 4 ft. 3 in. by 4 ft. 10 in., and 27 in. deep, although dimensions vary slightly according to the type of the installation. Cooling a car from 90 F outside to 70 F inside may be done at an approximate rate of 1 F per min. It probably would take ½ hr. to cool a car 20 F.

As the water is cooled by the energy of the steam from the locomotive, electric power generated on the train is confined to the operation of fans, pumps and lights.

The tank holds approximately 400 lb. of water. The unit is designed so that it can be lifted from the car for repairs.

Another System

In an air conditioning system which is applied to Pullman cars, the power is obtained from the car axle, connected to a 32-volt generator for the car lighting system. The compressor output reaches its maximum at a set train speed and maintains this output constantly.

Separately from this power supply, an electric motor is provided for standby service, obtaining its power from the yard or station lines while standing at the station or in the yard.

The portion of the refrigeration equipment which is under the car is mounted in one box consisting of compressor, condenser, receiver tank and cooling fan for the condenser. *F-12* is the refrigerant used.

A heating coil is also provided overhead adjacent to the cooling coil and is used for partial heating of the car during cold weather and tempering of the air used in the ventilation of the car during mild weather. The heat supply for this coil comes from the steam train line, and is reduced to vapor before entering the coil. Humidity is maintained by vapor exhausted from the radiator, automatically controlled.

Of the air which is used for cooling and heating the car, approximately 25 per cent is drawn from outside the car and filtered, and 75 per cent is recirculated. It is then blown through the cooling coil or heating coil, entering ducts running longitudinally of the car, and flowing from outlets in the various rooms or compartments.

A Self-Contained Axle-Driven System

A recent individual-car, all-electric air conditioning and refrigerating system for railway cars uses power from the axle. A storage battery enables the system to operate when the car is standing still as well as when it is in motion. At low train speeds sufficient power is generated to operate the system; at higher speeds the power is stored for use in precooling or when the train is standing still.

Air Travel: Air is drawn from the car into the air conditioning unit beneath the ceiling at one end of the car. Here it passes over frosted fin-surface cooling coils containing liquid *F-12*. Heat and moisture are absorbed by the coils and foreign matter is removed from the air by contact with the frosted surfaces. The air may be mixed in the air conditioning unit with air from the outside drawn through a grille in the ceiling of the vestibule and passing through a filter. The conditioned air is discharged into a duct along one side of the car in the space usually devoted to ventilators and is distributed throughout the car. The air is returned to the air conditioner through the return grille.

Refrigeration: A closed-cycle refrigerating unit consists of a compressor and motor beneath the car. The refrigerant in the form of vapor passes from the compressor to the liquefying unit which may be a cooling tower liquefier or an air-cooled liquefier. The liquefied refrigerant then passes through a float regulator to the fin coils in the air conditioner and is returned to the compressor, completing the cycle.

In the air-cooled liquefier, air is drawn into the chamber through a vent in the side of the car, blown over the coils and is discharged at the top of the car.

Controls: By means of a button control panel the system is started, operated with or without refrigeration, or stopped. If the battery should become under-charged a hand switch delivers to it the full power available. A thermostat stops all equipment but the air conditioner fan when a predetermined temperature is reached.

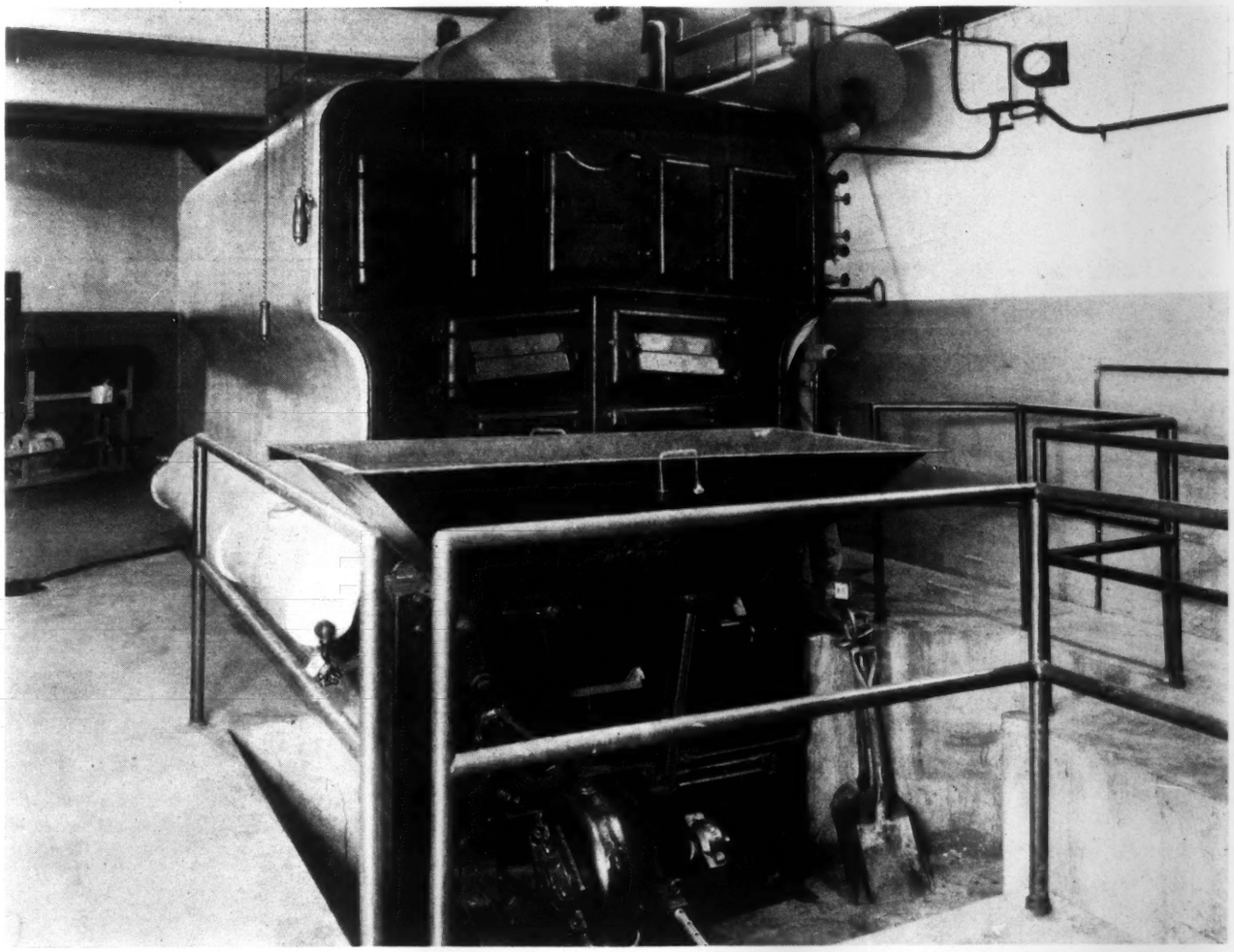


FIG. 8—STOKER AND BOILER INSTALLATION IN CHANDLER SUBSTATION OF THE DETROIT EDISON COMPANY

SELECTING THE RIGHT SIZE HEATING BOILER . . . .

By Sabin Crocker*

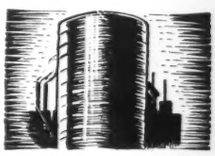
AMONG the items affecting the selection of the right size steam or hot water heating boiler are the following:

(*1*) Amount of connected (standing) radiation.

(*2*) Heat loss from bare and covered piping in supply and returns.

(*3*) Domestic hot water requirements.

(*4*) Satisfactory warming-up period from a cold or partially cold condition of the entire system.



(*5*) Efficiency with hard or soft coal, gas- or oil-firing as the case may be.

(*6*) Grate area with hand-fired coal, or fuel burning rate with stokers, oil or gas.

(*7*) Combustion space in the furnace.

(*8*) Type of heat liberation, whether continuous or intermittent or a combination of both.

In the February issue the first three of the above items were discussed in detail; last month Item *4* was considered, and this article is devoted to Items *5, 6, 7* and *8*.

* Engineer, The Detroit Edison Company, Detroit, Mich.

Among the items which must be considered in the selection of the right size heating boiler are the grate area with hand-fired coal, or the fuel burning rate with stokers, oil or gas; the combustion space in the furnace; and the type of heat liberation. How these factors are allowed for is told in this article. . . .

Fuel-Burning Rate

Maximum fuel-burning rate is an important item to consider in selecting a heating boiler. Inability to burn sufficient coal, oil or gas—as the case may be—will prolong the warming-up period and may result in failure to carry the designed radiation load. With a hand-fired coal job this difficulty can be overcome after a fashion by excessive forcing of the fire, provided sufficient draft is available. With a stoker, or with oil or gas firing, the maximum combustion rate is limited by the capacity of the fuel-burning equipment, and further forcing is impossible. It is imperative, therefore, to pay even more attention to securing ample capacity with such equipment than it is with manual firing of coal. In any event, it is good practice to select fuel-burning equipment which has sufficient reserve capacity to carry the maximum load with some margin to spare. In the following discussion of different types of firing it is assumed that a chimney is provided of adequate proportions to produce the requisite draft and carry away the products of combustion under maximum conditions.

Hand-Fired Coal Boilers

With hand-fired coal boilers the maximum combustion rate depends on the kind of fuel and the grate area. Higher combustion rates with a given fuel are expected in large boilers than in small ones. This is due in part to ability to burn more fuel per sq. ft. of grate area in a large boiler, as well as to shorter firing intervals and more skilled attendance. Residence boilers and other small jobs are expected to operate for comparatively long periods without attention, while larger boilers operated by a fireman can be tended more frequently. Table 12 gives the recommendations of the *A.S.H.V.E.* for practical maximum combustion rates for various fuels and ranges of grate area.

The *Steel Heating Boiler Institute* has established a code for the rating of steel boilers which relates output and grate area. According to this code, the sq. ft. of 240-Btu radiation represented by Items *1*, *2* and *3* (previously described) shall not be more than fourteen times the sq. ft. of heating surface for hand-fired boilers burning solid fuel. On the basis that this ratio is not exceeded, minimum grate area is determined from the following empirical formulas:

For boilers with ratings between 300 and 4,000 sq. ft. of steam radiation:

$$\text{Grate area} = \sqrt{\frac{(\text{Rating in sq. ft. of steam radiation}) - 200}{25.5}}$$

For boilers with ratings of 4,000 sq. ft. of steam radiation and larger:

$$\text{Grate area} = \sqrt{\frac{(\text{Rating in sq. ft. of steam radiation}) - 1{,}500}{16.8}}$$

Table 12—Practical Combustion Rates for Relatively Small Coal-Fired Heating Boilers Operating on Natural Draft of from ¼ In. to ½ In. Water[a]

Sq. Ft. Grate	Kind of Coal	Lb. of Coal Per Sq. Ft. Grate Per Hr.
Up to 4	No. 1 Buckwheat Anthracite	3
5 to 9	No. 1 Buckwheat Anthracite	3½
10 to 14	No. 1 Buckwheat Anthracite	4
15 to 19	No. 1 Buckwheat Anthracite	4½
20 to 25	No. 1 Buckwheat Anthracite	5
Up to 9	.....Anthracite Pea.....	5
10 to 19	.....Anthracite Pea.....	5½
20 to 25	.....Anthracite Pea.....	6
Up to 4	.Anthracite Nut and Larger.	5
5 to 9	.Anthracite Nut and Larger.	6
10 to 14	.Anthracite Nut and Larger.	7
15 to 19	.Anthracite Nut and Larger.	8
20 to 25	.Anthracite Nut and Larger.	9
Up to 4	.....Bituminous.....	4
5 to 14	.....Bituminous.....	7
15 and above	.....Bituminous.....	10

[a] Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq. ft. than those indicated in this table.

No such empirical rules have been established for cast-iron boilers, although the tendency is toward the publication of performance curves and catalog data regarding boiler heating surface and guaranteed efficiencies at various rates of heat transfer. Typical values for the relation between grate area and heating surface of steel and cast-iron boilers are compared in Table 13.

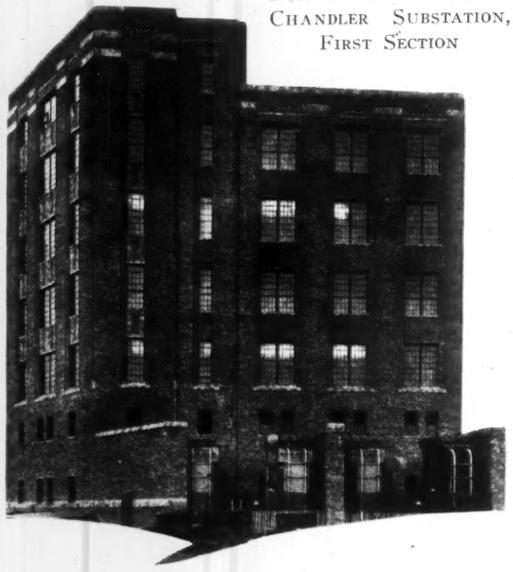


Fig. 9.—Exterior of Chandler Substation, First Section

Heating Surface

It will be noted that the tendency is to provide more heating surface for a given grate area in steel boilers than in cast iron. This would indicate that for the same rate of heat transfer, a higher rate of combustion per sq. ft. of grate area is assumed in the case of steel boilers. On the other hand, for a given rate of combustion per sq. ft. of grate area, more heating surface is provided in steel boilers. Also for the same rate of heat transfer per sq. ft. of surface, since more surface is provided, the steaming capacity of a steel boiler per sq. ft. of grate area is greater than that for a cast-iron boiler. *If selection is made on a basis of grate area,* this should result in higher efficiency in a steel boiler than in a cast-iron boiler *of the same grate area.* Another interesting condition brought out by Table 13 is the increase in heating surface per sq. ft. of grate area with increase in boiler size. This fits in with the higher combustion rates per sq. ft. of grate area in larger boilers and tends toward a uniform rate of heat transfer per sq. ft. of boiler surface throughout the range of boiler sizes.

The *Heating and Piping Contractors National Asso-*

ciation has adopted a set of engineering standards for the selection of cast-iron heating boilers. These standards relate connected load to grate width and boiler length in rectangular sectional boilers and extensive tables are given for the allowable load corresponding to various grate widths and boiler lengths. Similar data are given for round cast-iron sectional boilers in which credit is given for the extra convection surface obtained by adding more sections. These tables afford a practical basis of selection for conventional types of cast-iron boilers whose characteristics are comparatively uniform, but they would not be of value for special design such, for instance, as those having pin-type extended surface, or zig-zag water tubes.

Stoker-Fired Coal Boilers

Where stokers are installed in low - pressure heating boilers, common practice is to size the boiler as for hand firing using the customary heating surface and grate area called for by the load. The usual grates are removed, of course, and a suitable retort and dead plate installed in their place. The stoker should be selected with a view to obtaining the requisite coal-burning capacity by a safe margin. The blower used with the stoker must be able to supply approximately 3 1/3 c.f.m. of air (measured at 70 F) for each pound of coal to be burned per hour. Ample combustion space above the fuel bed is highly desirable to insure smokeless combustion, and heights of combustion chambers sometimes are fixed by state or municipal codes. Table 14 gives setting heights recommended for heating boilers.

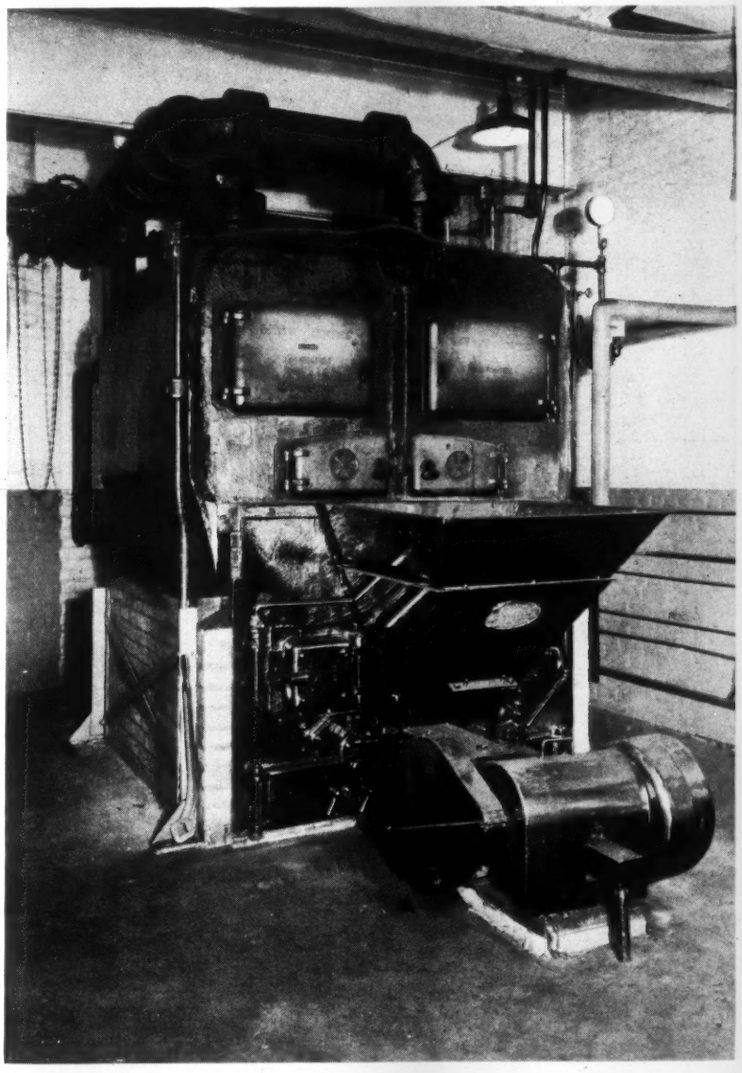


FIG. 10—STOKER AND BOILER INSTALLED IN HOWARD SUBSTATION

The actual loads referred to in Table 14 represent the sum of Items *1, 2* and *3* outlined in these articles, rather than boiler manufacturers' ratings. Furnace volumes are based on a heat liberation rate of 50,000 Btu per hr. per cu. ft. of combustion space with a view to insuring smokeless combustion with high volatile fuels. The furnace heights called for in Table 14 are not always realized in actual practice, especially when stokers have to be fitted in existing boilers with a minimum of rearrangement. The writer knows of several 36-in. cast-iron boilers which have been fitted with underfeed stokers where the tops of the retorts were installed at the former grate level and satisfactory operation obtained with a furnace height of not to exceed 26 inches. This is less by the lettered dimension *A* denoting the required depth of stoker pit or height of raised setting than the furnace height of 48 to 51 in. called for in the section of Table 14 designated "Compact Welded Boilers" which comes the closest to fitting this condition. Many compact welded boilers show a deficiency in this respect similar to that of cast-iron boilers, although not always to the same extent.

In order to get the required furnace height with cast-iron or compact welded boilers it usually is necessary either to pit the stoker, or, where water-line conditions and headroom permit, to raise the boiler on a brick foundation setting. The latter course has advantages, where

TABLE 13—TYPICAL VALUES FOR SQ. FT. OF BOILER HEATING SURFACE PER SQ. FT. OF GRATE AREA

Boiler Heating Surface Sq. Ft.	Sq. Ft. Heating Surface Per Sq. Ft. Grate Area	
	Cast-Iron Sectional	Steel Portable Fire-Box Type
15—25	10—12	.....
25—50	10—12	15—17
50—100	10—12	15—17
100—500	11—16	20—30
500—1000	16—25	25—35
1000—3000	.....	35—55

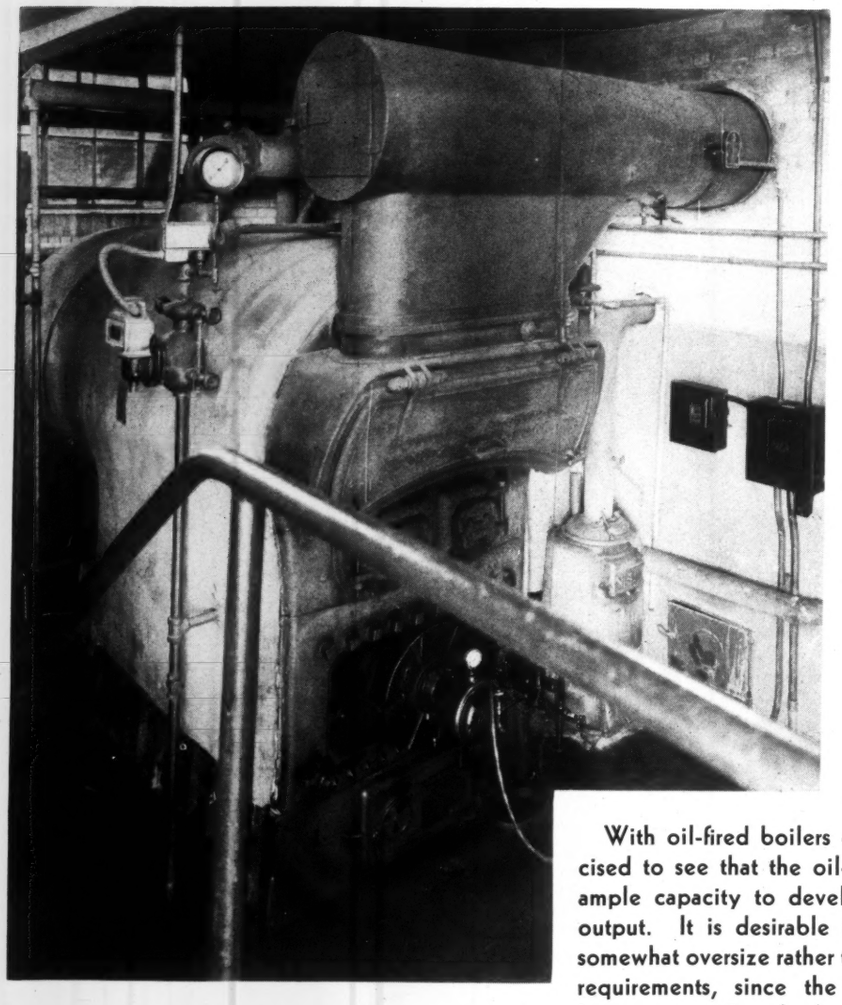


FIG. 11—OIL BURNER AND BOILER INSTALLED IN ECORSE ROAD WAREHOUSE

With oil-fired boilers due care must be exercised to see that the oil-burning equipment has ample capacity to develop the required boiler output. It is desirable to have the oil burner somewhat oversize rather than under the expected requirements, since the maximum rate of oil burning is definitely limited by the design of the burner. » » » » » » »

In selecting gas-fired boilers, it is the writer's opinion that it is better practice to select a somewhat larger boiler than the minimum and then throttle down gas and air adjustments as required to obtain optimum operating conditions.

FIG. 12—GAS BOILERS IN PORT HURON SALES OFFICE

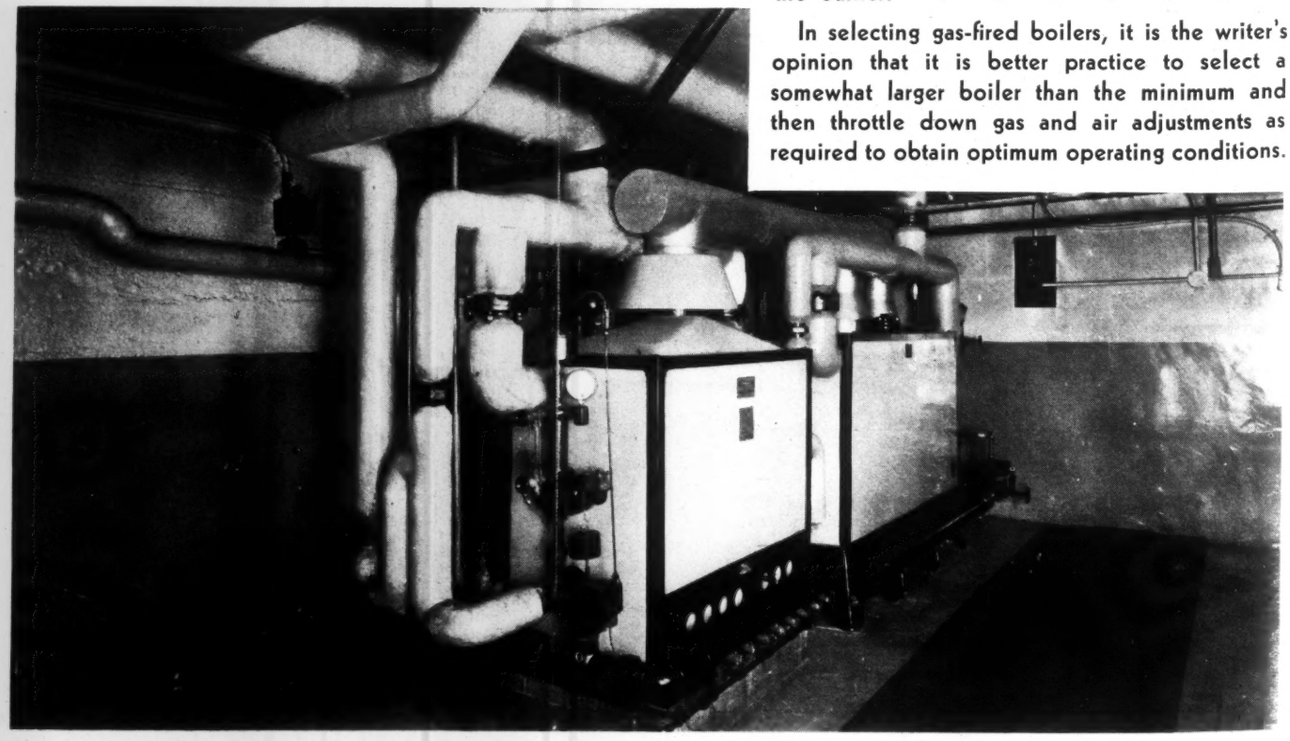


TABLE 14—RECOMMENDED SETTING HEIGHTS FOR HEATING BOILERS EQUIPPED WITH MECHANICAL STOKERS

Midwest Stoker Association

Firebox Boilers

ACTUAL LOAD	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	42″	48″	54″	60″	66″	72″	78″	84″	84″

A = Distance from bottom of Water Leg to floor.
B = Distance from Crown Sheet to bottom of Water Leg.

Compact Welded Boilers

ACTUAL LOAD	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	30″	33″	36″	42″	45″	48″	54″	60″	60″

A = Distance from bottom of Water Leg to floor.
B = Distance from Crown Sheet to bottom of Water Leg.

H. R. T. Boilers

HP.	50	75	100	125	150	175	200	225	250	275	300
A	5′-0″	5′-6″	6′-0″	6′-6″	7′-0″	7′-0″	7′-6″	8′-0″	8′-6″	9′-0″	9′-0″

A = Distance from bottom of shell to floor. HP. = Installed horsepower.

In the case of the FIREBOX or COMPACT WELDED type boilers the desired setting height can be obtained by combining the *A* and *B* dimensions. The load ratings shown for this class of boilers are actual developed loads in square feet of equivalent cast-iron steam radiation and are not manufacturers rating.

The setting heights given for H. R. T. boilers may be used for developed loads up to 50 per cent above normal rating.

The above tables were prepared by the *Midwest Stoker Association* as a guide for architects and consulting engineers in determining the head room requirements for heating plants.

practicable, in giving a better working space around the boiler and stoker. On the other hand where a monorail conveyor or other handy device is not provided for handling coal and ashes, it may be more convenient to fill the coal hopper of a pitted stoker than to shovel coal by hand into the hopper of a raised setting job. If the size of the system permits, it is desirable to have the hopper large enough to hold at least an eight-hour coal supply to carry the stoker through the night without attention.

A view of the pitted stoker installation for Chandler substation is shown in Fig. 8. The stoker in this instance is installed in a 54-in. header-type cast-iron boiler sized for carrying 13,000 sq. ft. of 240-Btu steam radiation in the ultimate building, the first section of which, representing a load of 8,841 sq. ft., is shown in Fig. 9.

A traveling hoist is used for handling coal and ash buckets in the installation shown in Fig. 8. This stoker has a single side dump with the clean-out door located alongside where the shovels are standing.

A double-dump stoker installed in a 50-in. push nipple cast-iron boiler mounted on a raised setting is shown in the view of the Howard substation boiler room, Fig. 10. The ultimate building in this case represents a load of 4200 sq. ft. of 240-Btu steam radiation. A monorail for handling coal and ashes also appears in this view.

Oil-Fired Boilers

With oil-fired boilers due care must be exercised to see that the oil-burning equipment has ample capacity to develop the required boiler output. It is desirable to have the oil burner somewhat oversize rather than under the expected requirements, since the maximum rate of oil burning is definitely limited by the design of the burner.

With a full-automatic burner it is desirable to use a larger boiler than would be required with coal firing, since the oil fire is operating at its maximum rate while on and adjusts itself to weather conditions by the length of time between periods of operation. This is quite different from coal firing where it is seldom necessary to push the fire to its maximum intensity, and most of the fuel used during the season is burned in a slow fire. Consequently a somewhat larger boiler is required with oil to obtain as low a stack temperature as would correspond to normal operation with coal.

With semi-automatic or with manually-operated oil burners it is not necessary to operate them at maximum output all the time, as the fire can be adjusted to suit load requirements. Under these conditions the necessity for a large boiler is not so evident, although it may be found desirable from a standpoint of fuel saving.

The amount of combustion space required for burning oil depends to a considerable extent on the design of the burner and furnace. A good general rule is that there should be one cubic foot of furnace volume for each 1½ to 2 lb. of oil burned per hr. There are successful oil burner and boiler combinations on the market, however, which do not require this furnace volume. The absence of the incandescent fuel bed existing with a solid-fuel fire lessens the necessity for large furnace volume and extensive radiant heating surface. Liberal provision of convection surface seems more important with oil and gas firing than does large furnace volume and the present tendency is for ample heating surface in the flue passages.

The *Steel Heating Boiler Institute* code previously referred to allows 17 sq. ft. of 240-Btu. radiation per sq. ft. of boiler heating surface where oil or gas is burned or where solid fuel is mechanically fired, as in stokers. This load is 21.5 per cent greater than the Institute's recommendation for hand-fired boilers burning solid fuel. While it may be possible to force boilers with oil, gas or stoker firing to this extent, it usually is not desirable with oil or gas from an economy standpoint. Consequently the general trend seems to be contrary to this particular recommendation.

Where an oil burner is installed with a refractory lining inside the combustion chamber, pains should be taken to see that the refractory setting does not screen an undue amount of the radiant heating surface in the furnace, as available radiant surface should be utilized to the fullest extent. Many cast-iron boilers are built without base sections and have water legs coming down to and resting on the floor. This water leg surface has a high rate of heat absorption if exposed to direct radiation from the oil flame, and should not be screened more than necessary. For this reason such boilers should be installed on a raised setting, or the burner pitted sufficiently to keep as much of the refractories as possible below the water legs. A stoker-fired boiler without base section and mounted on a raised setting as proposed in this instance is shown in Fig. 10. The setting need not be so high for oil burning since oil can be burned satisfactorily with less furnace volume than is desirable for the smokeless combustion of high volatile coal. The writer knows of several instances where the performance of oil-burning boilers was greatly improved by changing

the setting in this way. Usually with existing installations this is most easily accomplished by pitting the burner.

A semi-automatic oil burner installed in the Ecorse Road warehouse is shown in Fig. 11. The boiler is of the return-tubular fire-box type with 297 sq. ft. of heating surface and has a total connected load of 3,016 sq. ft. of 240-Btu radiation. This represents 10.1 sq. ft. of radiation per sq. ft. of boiler heating surface and performance has been very satisfactory. The burner is of the centrifugal-atomizing-type, manually started and with a high- and low-fire control operated by a steam pressure bellows. The combustion rate varies from 3 to 15 gal. per hr. One of the special features of this job is provision for incinerating trash on water tube grates which are the usual type provided in down-draft smokeless coal-burning boilers. This arrangement is made possible by mounting the burner entirely outside the boiler on the lower firing door, thus leaving the combustion space clear and the ash-pit door free for the removal of refuse.

Gas-Fired Boilers

Selection of the right size gas boiler is discussed at some length in the bulletin on house heating, (third edition) of the *American Gas Association*, from which the following is quoted:

"Unlike boilers designed to burn solid fuel, the question of the firing period or available fuel capacity is not a factor in estimating the size of gas boilers, which instantly deliver their full rated capacity whenever the gas is turned on. The highest operating efficiency is obtained when the rated output of a gas boiler most nearly equals the exact heating requirements of the building.

"It has been the common practice to assume that the extra tax imposed on the boiler due to piping installed is about 25 per cent of the amount of column radiation and to add a further 25 per cent to this total to take care of starting load or "pick up" under thermostatic control.

"The proportionate amount of piping to radiation is considerably greater in a small than in a large installation, and while 25 per cent in general instances fairly represents the tax imposed, yet upon a percentage basis the additional load imposed on the boiler by the amount of piping necessary in a small installation may be as high as 40 per cent of the amount of radiation installed, whereas on very large jobs this percentage may be as low as 15 per cent.

"A gas-fired boiler is so sensitive to water temperature or steam pressure regulation, as well as automatic room temperature control, that in most cases adding 25 per cent for "pick-up" load will be ample for average continuous residence heating. Where the building is irregularly or periodically heated, either by automatic or hand control, this percentage should be increased to 50 per cent or 60 per cent.

"The most accurate method for selecting the proper size of boiler for any given installation is to determine the exact amount of standard column radiation required to heat the building to the temperatures required, and then to add the actual heat losses from covered or uncovered piping in units of sq. ft. of standard column radiation.

"Table 15 gives the factors by which the installed radiation should be multiplied to determine the size of boiler required. This is designed to take care of starting and piping allowances for systems under automatic control. The table can only be used, however, where the installation is all direct radiation designed for interior temperatures at or near 70 F. This method of estimating the excess capacity required is not suitable for office buildings, garages, stores, etc. It is essentially a system for private residences.

"The boiler is selected then by adding the piping and starting allowance to the installed radiation, either in terms of sq. ft. or Btu. From the manufacturer's catalog select the boiler with nearest rating larger than the estimated requirement.

"These ratings can be determined in accordance with the methods of the *A. G. A.* test code or the manufacturer's rating. No definite figures on the rating per burner or per boiler section can be given since so many types and designs are available.

"In the event that the owner does not wish to install a thermostatically-controlled system, the procedure is somewhat different. In place of adding 25 per cent for risers and returns and 25 per cent for quick start, it will be found that, unless the boiler is selected on the basis of a 100 per cent allowance for manual control, the operation will not be satisfactory. The 100 per cent includes the allowance for risers and returns, no separate addition being necessary in this case. With manual control, the temperature fluctuations within the building are quite considerable because the reaction of the human body to temperature change is not nearly so sensitive as that of a thermostat. A greater amount of boiler capacity is therefore necessary to effect a quick return to comfortable temperatures."

Table 15—Gas-Fired Boiler Selection Factors

Multiply Installed Radiation by Factor to Find Minimum Boiler Size

Installed Radiation (Either Steam or Hot Water)	Factor	Installed Radiation (Either Steam or Hot Water)	Factor
100	1.56	2200	1.445
200	1.56	2300	1.443
300	1.56	2400	1.439
400	1.56	2500	1.437
500	1.56	2600	1.435
600	1.552	2700	1.432
700	1.545	2800	1.43
800	1.537	2900	1.427
900	1.53	3000	1.424
1000	1.523	3100	1.422
1100	1.515	3200	1.42
1200	1.508	3300	1.417
1300	1.501	3400	1.414
1400	1.494	3500	1.412
1500	1.486	3600	1.409
1600	1.478	3700	1.407
1700	1.472	3800	1.405
1800	1.464	3900	1.402
1900	1.457	4000 and over	1.40
2000	1.45		
2100	1.447		

Manufacturers of gas-heating boilers usually rate their products in terms of *A. G. A.* steam or water radiation (240 or 150 Btu respectively), with instructions that the load be computed as described above and a boiler selected which has a rating equal to or slightly greater than the connected load plus piping tax and warming-up allowance. Where gas boilers are used the domestic water supply commonly is heated in separate gas water heaters, under which circumstances it is not necessary to add the water heater load in figuring boiler capacity.

In selecting a gas boiler by the *A. G. A.* rating method it should be borne in mind that this procedure gives the minimum size boiler which can carry the load, and that output is limited by gas-burning capacity as well as by boiler heating surface. In the writer's opinion it is

better practice to select a somewhat larger boiler and then throttle down the gas and air adjustments as required to obtain optimum operating conditions. This will tend to give a low stack temperature with high efficiency, and at the same time provide reserve capacity in case the job was under-estimated or more load is added in the future. Provision of some capacity beyond the absolute minimum recommended for automatic control is desirable also to furnish quick pick-up if the building temperature is lowered at night as is usually the case with gas firing. The writer believes, therefore, that it is advisable with automatic control as well as with manual to follow the instructions given in the last paragraph quoted from the *A. G. A.* bulletin and add at least 100 per cent to the standing radiation to provide properly for piping tax and pick-up. This is borne out by the instance given below.

The dual gas-boiler installation in the Port Huron sales office is shown in Fig. 12. When alterations were required in the heating system serving the sales office and the meter department located in an adjacent building it was decided to use gas. The combined load, including radiation and piping tax, amounted to 2,175 sq. ft. of 240-Btu radiation.

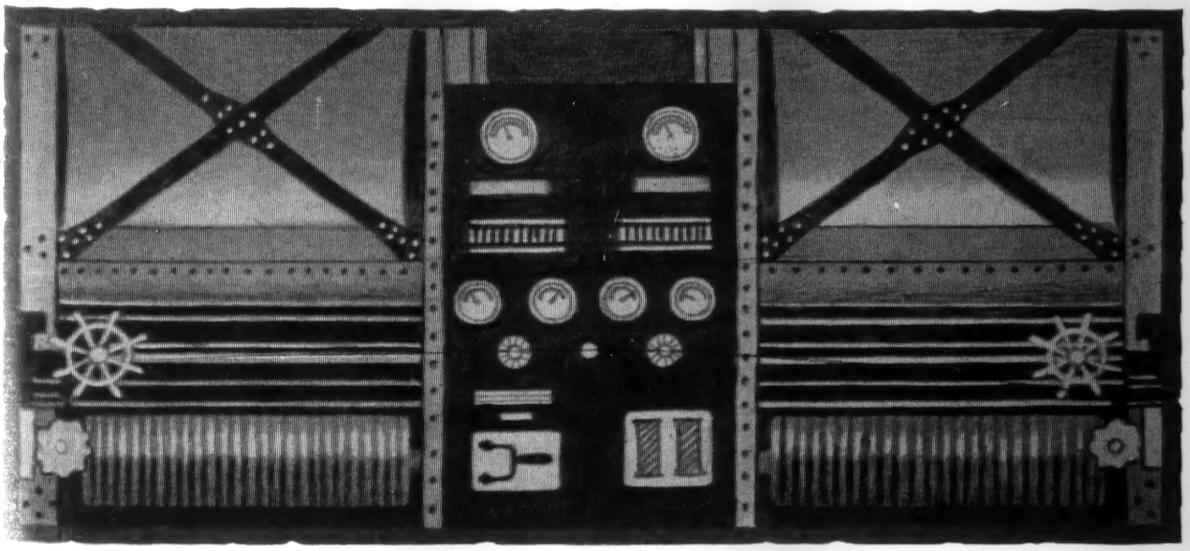
Distribution of the load in the two buildings is as follows:

	Sq. Ft. of 240-Btu radiation	
	Sales Office	*Meter Dept.*
Standing radiation	1,022	737
Piping tax	205	211
Total	1,227	948

It was decided to split up into two separate systems and install two gas boilers with full automatic control. This gave opportunity to select two makes of boilers and observe their relative performance, including different types of controls. The vendor selling the meter department boiler was inclined to be liberal in his selection and recommended a boiler having 86 sq. ft. of heating surface and an *A. G. A.* rating of 1,500 sq. ft. of 240-Btu radiation. Referring to the 737 sq. ft. of standing radiation, the *A. G. A.* rating represented an allowance of 104 per cent for piping tax and pick-up. Referring to the total load of 948 sq. ft., there were 11 sq. ft. of 240-Btu radiation per sq. ft. of boiler heating surface. This boiler at once demonstrated its ability to get up steam pressure on short notice and has given good satisfaction in every respect.

The vendor furnishing the sales office boiler was optimistic as to the steam-generating ability of his boiler and the significance of *A. G. A.* ratings and recommended an eleven-section boiler listed as having 137.5 sq. ft. of heating surface and an *A. G. A.* rating of 1,540 sq. ft. of 240-Btu radiation. Referring to the 1,022 sq. ft. of standing radiation the *A. G. A.* rating represented an allowance of 50 per cent for piping tax and pick-up. Referring to the total load of 1,227 sq. ft. there were 9 sq. ft. of 240-Btu radiation per sq. ft. of boiler heating surface. It will be noted that the relation between boiler heating surface and *A. G. A.* rating varies to a marked degree between these two boilers, probably due to the different types of extended surface and flue gas passages. With special types of boilers of this kind it evidently is safer to depend on *A. G. A.* ratings or other performance data than on the amount of heating surface. The eleven-section boiler was unable to get up steam pressure during a prolonged test. The radiators eventually became warm, but no appreciable steam pressure was developed. This condition was deemed unsatisfactory and the vendor was requested to enlarge the boiler to seventeen sections with 212.5 sq. ft. of heating surface and an *A.G.A.* rating of 2,380 sq. ft. Referring again to the 1,022 sq. ft. of standing radiation the *A. G. A.* rating represented an allowance of 132 per cent for piping tax and pick-up. The enlarged boiler was able to get up steam pressure in a short time and has performed satisfactorily.

The gas consumptions of both boilers were checked through for an entire heating season and found to be in substantial agreement with the *A. G. A.* methods of estimating gas consumption on a degree-day basis. The writer considers the foregoing examples as rather convincing proof of the desirability of selecting gas boilers of liberal size for the connected load.



Soot Blower Makes a 14½-% Fuel Saving

By G. L. Davis*

A FUEL saving of 14½ per cent has been made by the Belcrest hotel, Detroit, through the installation of an air-operated mechanical soot blower which cleans soot and ash automatically from the flues of its low-pressure fire-tube heating boiler.

The equipment consists of a starting switch, motor-driven air compressor, air receiver, blower unit and operating ratchet; the blower operation is entirely automatic. When the starting switch is thrown by means of a push button, the blower is cut in and the compressor begins operating, pumping air into the receiver until the pressure reaches 125 lb. At this point the pilot valve *12* is actuated, opening the main line valve *9* and admitting air to the blower *14*. The blower operating ratchet moves the blower arm through the cleaning range in arcs of 10 degrees. During this movement the air pressure in the receiver drops from 125 lb. to 75 lb., at which latter figure the blower cuts out. It then requires about three minutes for the compressor to again build up pressure to 125 lb., during which time the blower is stationary. At 125-lb. pressure, the cycle begins again and continues until the full 360-degree travel has been completed.

In addition to the fuel saving made intangible savings in labor, ability to carry the load readily and ability to maintain uniform heat throughout the building have resulted.

*Diamond Power Specialty Corporation, Detroit, Mich.

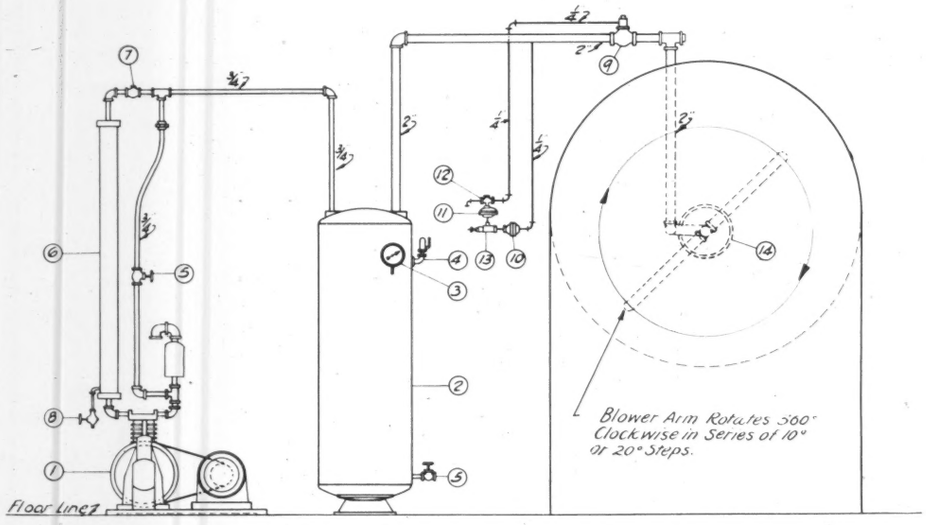


DIAGRAM SHOWING THE OPERATION OF THE AIR-OPERATED MECHANICAL SOOT BLOWER.

1, compressor unit (compressor, motor, air cleaner and starting switch; *2*, air receiver; *3*, 250-lb. air gage; *4*, 1-in. pop valve; *5*, ¾-in. globe valve; *6*, exhaust muffler; *7*, ¾-in. check valve; *8*, ¼-in. drain cock; *9*, 2-in. piston-operated valve; *10*, air screen; *11*, diaphragm; *12*, ⅜-in. whistle valve; *13*, unloading valve; *14*, soot blower.

Bridgman's Book on High Pressure Reviewed

The Physics of High Pressure, by P. W. Bridgman, professor of physics, Harvard University. The Macmillan Company, 1931. Price $5.00. Reviewed by F. E. Wertheim.

Dr. Bridgman, who has for 25 years investigated phenomena associated with high pressure, has written a book which will appeal to engineers as well as to physicists.

He has successfully attained the use of pressures as high as 40,000 kg. per sq. cm.,* for which he had to originate equipment to produce such pressures and gages to measure them accurately. It is interesting to note that the success of these projects in experimental science was based upon "The development of a packing technique which made it possible to reach, without leak, any pressure allowed by the mechanical strength of the containing vessels." The largest tubes used in this laboratory apparatus are ⅛ in. I.D. and 1¼ in. O.D., drilled from solid rods of alloy steel and heat treated.

Piping for engineering or industrial uses is subjected to very low pressures in comparison to the foregoing, but it must be remembered that much larger sizes are dealt with and, unlike the laboratory apparatus, creep stresses due to high temperatures may also be a factor. Notwithstanding the differing conditions, the book will yield information of practical value in piping design.

Among the principal chapters are: The measurement of high pressures, rupture phenomena, *p-v-t* relations in fluids, on compressibility in solids, on melting phenomena, on electrical resistance of metals and solids, on viscosity, and on other effects of high pressure.

The literature of physical science contains very few books covering this field of research. Dr. Bridgman's contribution is an outstanding achievement.

* 550,000 lb. per sq. in.

Air Conditioning Needs *of* Hospitals Specified

[*By* Dr. C. A. Mills*]

SINCE new developments in hospital construction engineering render it feasible to provide proper air conditioning at low cost, it is necessary that the medical profession indicate the need for this service and the use to which it may be put. This however, is a relatively new field and few physicians have given thought to the benefits that might be obtained for the hospital patient through proper artificial control of his environment. It is the writer's object in this brief article to suggest a few of the ends that might be obtained by air conditioning. Experience with patients of all types in properly conditioned hospitals may indicate much more extensive benefits than are suggested by our present meager knowledge.

Deserving of first mention is its use for both premature and full term infants of delicate constitution. The rigors of life, with extremes in both directions, should be kept away from these frail individuals with all possible care. Even infants of normal health should have a well regulated environment.

Next come the children suffering from summer diarrhea. Here relief from the heat, which seems to be a major factor in causing the disturbance, is of the utmost therapeutic importance. It has been shown statistically that there exists a close correlation between summer diarrhea and the heat waves of summer.

It would be of like importance to relieve heat stroke and heart failure cases from the depressive effects of hot summer weather. For these individuals the direct application of cold to the body, as with cold packs, is not nearly so much to be desired as that they have cold air to breathe. They should be provided with a room in which frequent and fairly wide fluctuations of temperature could be secured at will.

Surgeons avoid as many serious operations during the summer heat waves as they can because experience has shown that post-operative risks are unduly high. Relief from the summer heat is, therefore, not just a matter of comfort for the ill patient, but often becomes a matter of vital importance. The writer believes every surgical ward should be equipped to avoid temperature extremes and to provide a 10-12 F day and night variation. The temperature variability should be kept low for the first two or three days after operation, but should then be rapidly increased to provide maximum stimulation and shorten convalescence. This would hold as well for medical, obstetrical and other types of patients as for those in the surgical wards.

Air conditioning also should include the furnishing of certain rooms in each hospital with a high, even temperature and high humidity for the treatment of those conditions of over-stimulation, such as exophthalmic goiter or hyperthyroidism; severe cases of diabetes and pernicious anemia, essential hypertension, and many hyper-irritable or excitable nervous states. There has been practically no use made of climatic treatment in these different conditions but the betterment which they usually experience in the hot summer weather and during sojourns to regions of even high temperature warrants its use in hospitals in the more acute or severe cases. A week or two at a temperature of 85 F and a relative humidity of 70 per cent produces quite a noticeable reduction in the rate of body metabolism.

In general we have found experimentally that the body is best stimulated by fairly wide daily fluctuations of the temperature, and is most depressed by a constantly high temperature and high humidity. Since most of the patients' time in the hospital is spent recuperating, we should be able to provide the optimal environment for hastening this recovery. Either stimulation or depression should be applied as necessary.

Briefly listed, a few of the hospital's needs for air conditioning are:

1. For infants, especially the premature and delicate.
2. For children with summer diarrhea and other debilitating conditions.
3. For patients suffering from heat stroke or exhaustion, heart failure, etc.
4. For surgical patients to obviate the deleterious effects of summer heat.
5. For bronchitis and asthma—a dust-free, dry air.
6. Humid warmth for such over-stimulative diseases as hyperthyroidism, essential hypertension, acute or severe cases of diabetes and pernicious anemia. Also for excitable nervous states.

*Dept. of Internal Medicine, Cincinnati General Hospital, Cincinnati, Ohio.

Calculations for Drying

In this article, the author shows how the amount of heat required for drying a product is calculated. Two typical examples are worked out to indicate the manner in which such computations may be made. To meet the need for solving high-temperature drying problems easily, Mr. Tomlinson has prepared four charts which are published here as Figs. 4 to 7.

By Malcolm Tomlinson*

SINCE factors such as fuel needs, boiler capacities and pipe sizes must, at least, be approximated as soon as possible in the roughest drying calculations it is necessary to determine heat losses and gains. The temperature of the air used, whether the system contemplated is to have gravity or mechanical air circulation, must be raised by the application of heat from some external source. If this air entering the dryer is humid more heat will be required than for dry air since more water vapor must be warmed. Heat also must be applied to raise the temperature of the substance or product to be dried and also the dryer itself. Few dryers and dryer rooms are perfectly insulated and a percentage of the heat supplied will be wasted. Another source of heat loss will be through cracks around doors and windows. This heating loss may be reduced by the heat supplied by employes, motors, lighting equipment and other heat-producing sources. When these losses and gains are finally balanced for summer and winter weather conditions totals are obtained which represent the minimum and maximum demands, respectively, on the heat-producing plant.

Specific Heat of the Material

Every substance may be heated one degree by a definite amount of heat per pound of the material. This factor is known as the *specific heat*. Values for the mean specific heat of many gases, liquids and solids will be found in the *A. S. H. V. E. Guide*. Additional values have been compiled and are given in Table 1. If the weight of the product, or of any substance in the product, is known, the exact amount of heat needed to raise the temperature of the product the required number of degrees can be obtained by multiplying weight times specific heat times temperature difference.

Calculations for Drying a Typical Product

Drying 3,000 repeating coils presents a special problem, as a very low moisture content must be secured to obtain required electrical characteristics. These particular coils consist of 1-lb. iron cores wound

TABLE 1—MEAN SPECIFIC HEATS OF SEVERAL MATERIALS

MATERIAL	BTU PER F	TEMP. RANGE
SOLIDS		
Aluminum	0.222	68 to 212
Carbon	0.241	32
Copper	0.092	62
Iron (pure)	0.116	32
	0.109	59
Lead(cast)	0.023	59
Tin	0.055	32 to 212
Zinc	0.091	64
Brick	0.22	32 to 212
Silicon	0.183	135
Sodium	0.29	.....
Steel (mild)	0.165	.....
Lead Oxide	0.055	32 to 212
Magnesia	0.22	32 to 212
Mica	0.206	68 to 208
Rubber (India)	0.27 to 0.48	32 to 212
LIQUIDS		
Alcohol—ethyl	0.648	104
Alcohol—methyl	0.59	41 to 50
Sodium chloride (salt)	0.776 to 1.003	15 to 140
Calcium chloride	0.65 to 0.79	104
Copper sulphate	0.848	53½ to 59
Ether—ethyl	0.529	32
Potassium hydrate	0.876	64½
Sodium hydrate	0.942	64½
Zinc sulphate	0.842	68 to 125½

* Consulting engineer, West Chester, Pa.

with 1 lb. of copper and insulated with ⅛-lb. of cotton impregnated with 1/6-lb. of oil. Assume the entering air is at 50 per cent relative humidity and 70 F. A moisture content for the cotton of the dried coil is desired of not more than 0.05 per cent of the bone-dry weight.

From cotton regain data the moisture content for the entering air condition is 5½ per cent. The difference in per cent multiplied by the weight of the cotton per coil and the number of coils gives 20.4 lb. of moisture to be removed. The same data show that the maximum drying temperature for atmo-

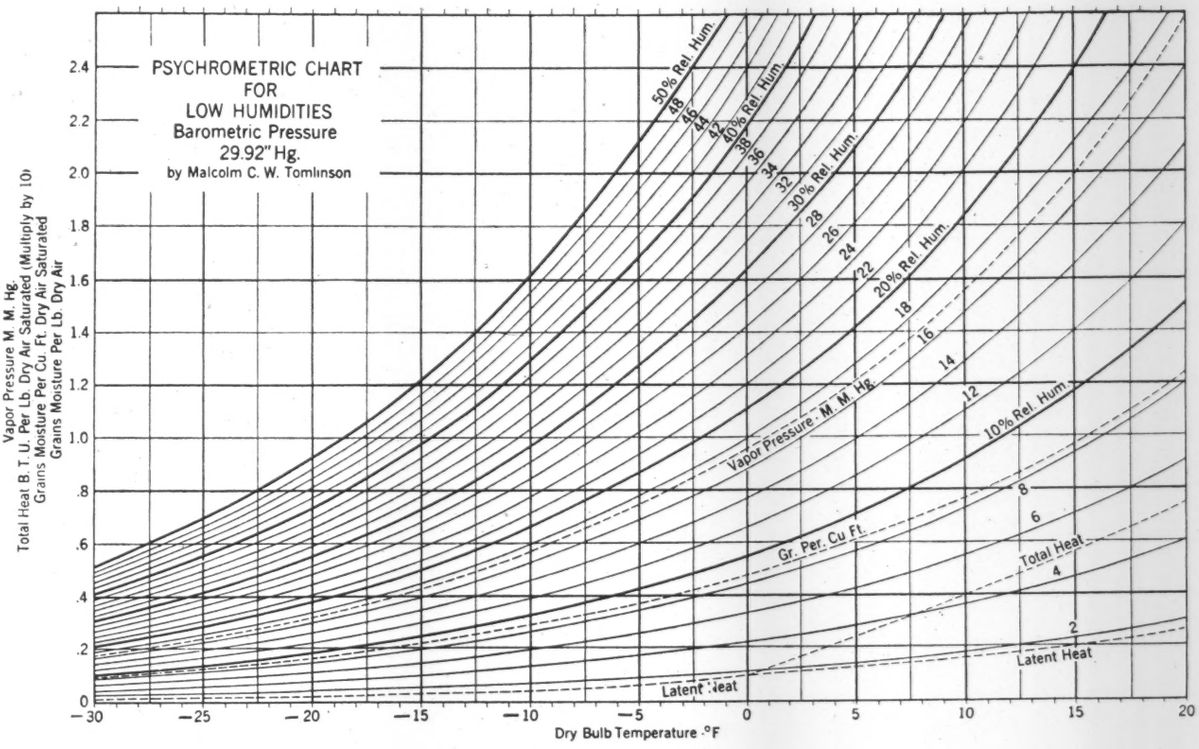


FIG. 1—LOW HUMIDITY CHART

Transactions, *American Society of Heating and Ventilating Engineers*, Vol. 35, 1929.

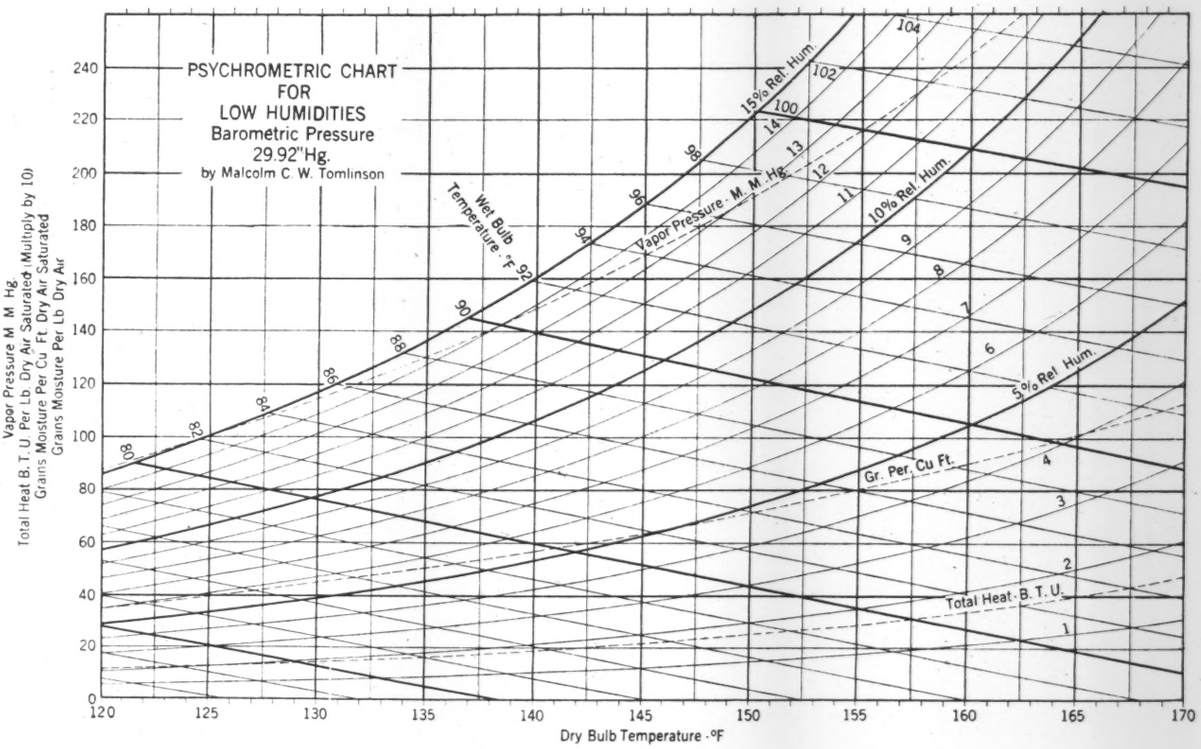

PSYCHROMETRIC CHART
FOR
LOW HUMIDITIES
Barometric Pressure
29.92" Hg.
by Malcolm C. W. Tomlinson
Vapor Pressure M. M. Hg.
Total Heat B. T. U. Per Lb. Dry Air Saturated (Multiply by 10)
Grains Moisture Per Cu. Ft. Dry Air Saturated
Grains Moisture Per Lb. Dry Air
Wet Bulb Temperature - °F
15% Rel. Hum.
10% Rel. Hum.
5% Rel. Hum.
Vapor Pressure - M. M. Hg.
Gr. Per. Cu Ft.
Total Heat - B. T. U.
Dry Bulb Temperature - °F

FIG. 2—LOW HUMIDITY CHART

Transactions, *American Society of Heating and Ventilating Engineers*, Vol. 35, 1929.

spheric pressure, which is 180 F, will provide the low moisture content at a relative humidity of ¼ per cent. Equivalent conditions can be secured by a 300-F temperature in a vacuum of 1-in. absolute.

With the low humidity psychrometric charts, the moisture contents of the entering and dried air are found as 0.0079 and 0.0014, respectively and the wet bulb is 82 F. Also, total heat in the entering air is 25.2 and in the leaving air, 45.2. The difference between the two moisture contents, or 0.0065, divided into the total weight of 20.4 gives 3,138 lb. of dry air which must be circulated; the difference in total heat, or 20 Btu per lb. of dry air, multiplied by the total weight of air, gives a total heat requirement of 62,760 Btu. This heat is for air alone. Heat for the coils can be calculated from the specific heats of the materials used and will be found to be 69,750 Btu. Neglecting other heat losses or gains the total heat which must be provided is 132,510 Btu.

The solution offered is for a simple type of dryer and is based on drying almost to the wet bulb temperature. In more complex types of dryers, counter air currents can be employed to reduce the time of drying. Here the temperatures and also the relative humidities of the entering and emergent air will gradually approach identical values.

A unique method of drying these coils by atmospheric pressure would be to remove the moisture from the entering and emergent air to a very low level by means of adsorption.

Product Must Be Sealed Against Moisture Regain

In all drying problems where the moisture content is reduced to very minute quantities and high temperatures are employed it is essential to seal the product against moisture regain before it is removed to the ordinary shop atmosphere. Some form of room must be provided outside of the dryer or the drying room where the air is held at a predetermined

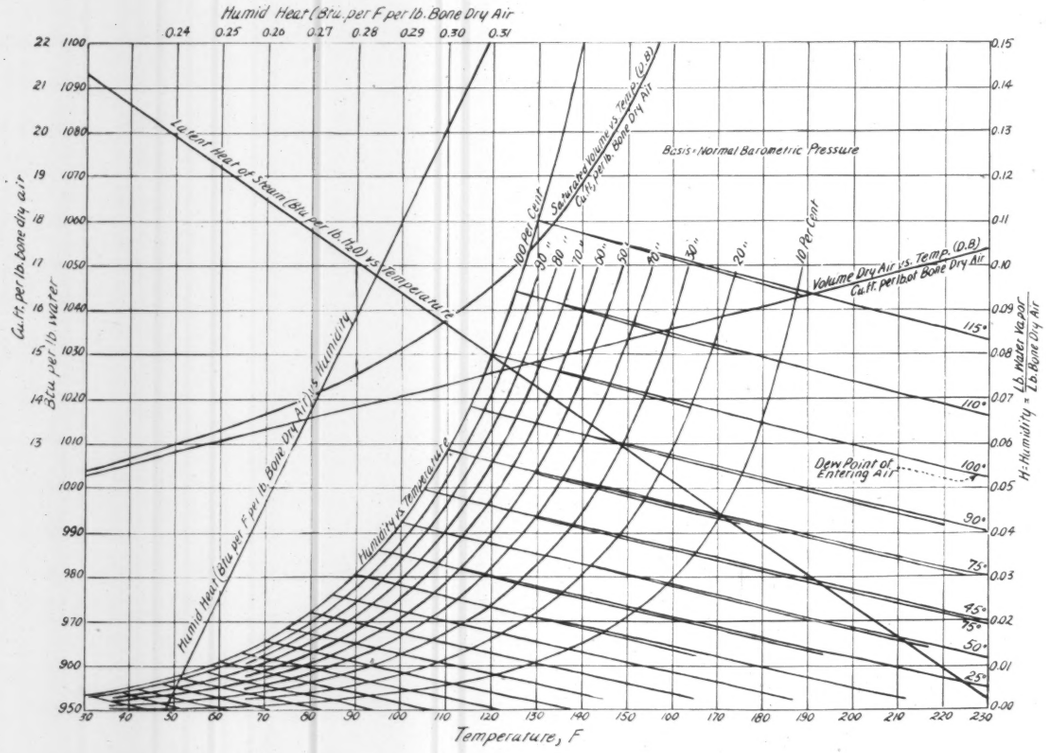

Humid Heat (Btu. per F per lb. Bone Dry Air
Latent Heat of Steam (Btu per lb. H2O) vs. Temperature
Basis: Normal Barometric Pressure
Volume Dry Air vs. Temp. (D.B.)
Cu. ft. per lbot Bone Dry Air
Dew Point of Entering Air
Humidity vs. Temperature
Humid Heat (Btu. per F per lb. Bone Dry Air) vs. Humidity
Cu. ft. per lb. Bone dry air
Btu. per lb. water
H: Humidity = Lb. water Vapor / Lb. Bone Dry Air
Temperature, F

FIG. 3—GROSVENOR PSYCHROMETRIC CHART, BASED ON "PER CENT HUMIDITY" RATHER THAN "PER CENT RELATIVE HUMIDITY"*

*The Units Used Are Defined as Follows:

Humidity—Pounds of water vapor carried by one pound of dry air. (The absolute humidity.)

Per Cent Humidity—Pounds of water vapor carried by one pound of dry air at a definite temperature divided by the pounds of vapor which one pound of completely saturated dry air will carry at the same temperature.

Humid Heat—Quantity of heat necessary to raise the temperature of one pound of dry air, plus such water vapor as it contains, one degree Fahrenheit.

Humid Volume—Volume of one pound of dry air, with the water vapor it contains, in cubic feet.

Saturated Volume—Humid volume at saturation.

As per cent relative humidity is defined as the ratio of the vapor pressure of the unsaturated mixture to the vapor pressure of the saturated mixture at the same temperature it is evident that the two relationships do not correspond.

The Grosvenor chart is based on "per cent humidity," while the charts in Figs. 1 and 2 are based on per cent relative humidity. To convert per cent relative humidity into per cent humidity multiply by the following factor:

$$\frac{760-p_s}{760-p}$$

in which p_s and p are the vapor pressures, respectively, of the saturated air and the air which is unsaturated at the same temperature. These pressures are in millimeters.

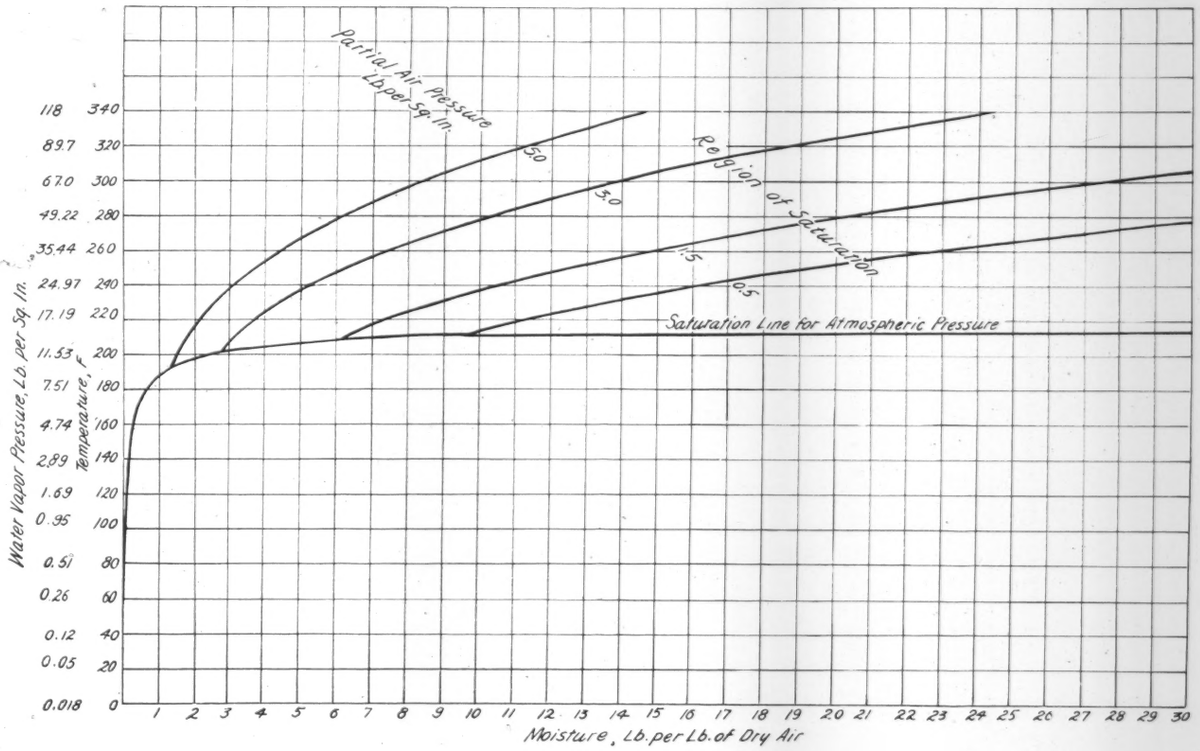


FIG. 4—MOISTURE IN AIR-WATER VAPOR MIXTURES AT HIGH TEMPERATURES

point of dryness but where the air temperatures are such that human beings can work in comfort.

Harmful Effect of Temperature Depends on Time

In drying vegetables there are many examples of the use of temperatures close to or above 212 F. Where time is a vital consideration and sufficient moisture can be removed quickly this plan has much to commend it for consideration. The harmful effect of temperature on the coloring or the physical structure of materials is a function of time. For example, a paper which will not be affected by 300 F at 1 lb. absolute for 12 hours would be partially carbonized if held under the same conditions 24 hours. In some paper mills temperatures around 400 F are used for rapidly drying moving sheets.

Drying Beets in Three Stages

Sugar beets might be selected as an example of high-temperature drying under so called atmospheric pressures. Here the moisture must be removed by decreasing a content of 76 per cent to 6 per cent, without loss or deterioration of the sugar content. Assume a conveyor-type of dryer to be used, which consists of three successive and separate stages.

In the first stage 65 per cent of moisture is removed at 250 F, in the second 25 per cent is removed at 230 F and in the third 10 per cent of the moisture is removed at 190 F. The drying cycle must not exceed one hour, the beets must travel in a direction opposite to the air and the humid air must leave the dryer at a temperature range of not more than 110 to 120 F in practically a saturated condition to obtain a high dryer efficiency. The weight of the fresh beets to be dried per hour is 7,500 lb.; of this weight 5,250 lb. represents the moisture to be removed per hour. Entering air is at 70 F and 40 per cent relative humidity.

First stage: In the first stage 3,412½ lb. of moisture must be removed at 250 F. If we use the Grosvenor psychrometric chart (Fig. 3) for the solution of this problem we can easily estimate from it that the heat to evaporate 1 lb. of water at 250 F is 939 Btu. To raise the temperature of the water to 250 F from 70 F requires 180 Btu. Thus a total of 1,119 Btu per lb. of water evaporated must be used or a total of 1,119 x 3,412½ or 3,818,600 Btu to actually warm the air before it enters the first stage. The humidity of the entering air is 0.006 lb. per lb. dry air and the humid heat of this air is 0.241 Btu. Therefore, the heat to raise 1 lb. of dry air through 180 F is 180 x 0.241 or 43.38 Btu. If we divide this quantity into the total heat we obtain 85,721.5 lb. of dry air. Again if the total moisture to be removed is divided by the lb. of dry air moved it is found that 0.04 lb. of moisture is removed from the beets per lb. of dry air. Adding to this the moisture in the entering air, or the humidity of entering air, it is found that the humidity of the air as it leaves the first stage of the dryer is 0.046 lb. per lb. of dry air.

Second Stage: Before the air from the first stage enters the second stage a portion is discarded. The temperature of the second stage is 230 F, the amount of moisture to be removed from the beets is 1,312½ lb. and the humidity of the entering air is 0.046 lb. By calculation the humid heat of this air is 0.238 plus 0.48 x 0.046 or 0.260. From the chart the heat to evaporate 1 lb. of water at 230 F is 953 Btu. If we multiply this latter quantity by the weight of mois-

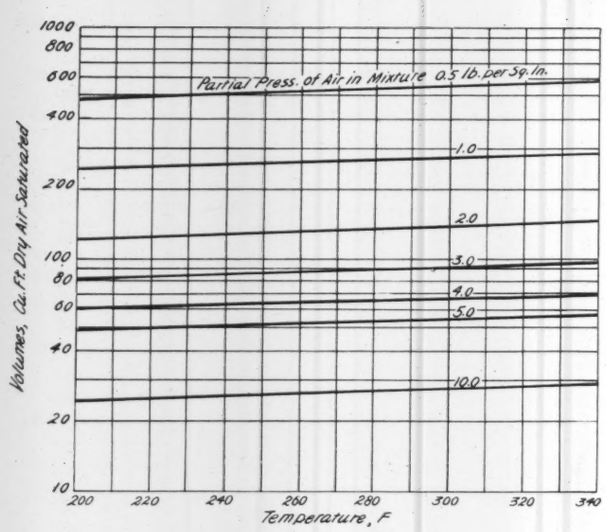


FIG. 5—VOLUMES OF SATURATED AIR-WATER VAPORS

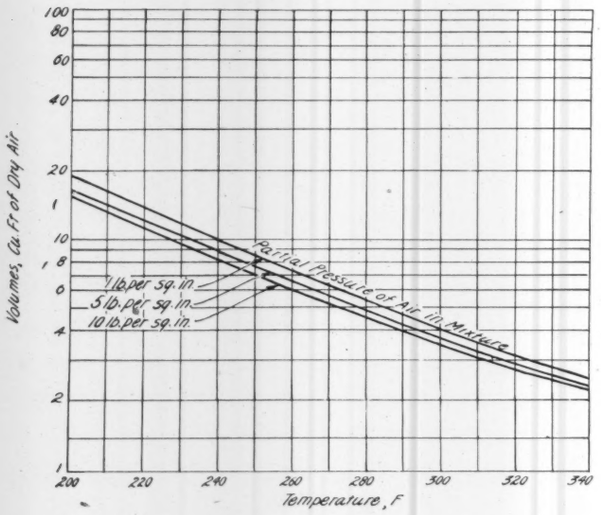


FIG. 6—VOLUMES OF DRY AIR-WATER VAPORS

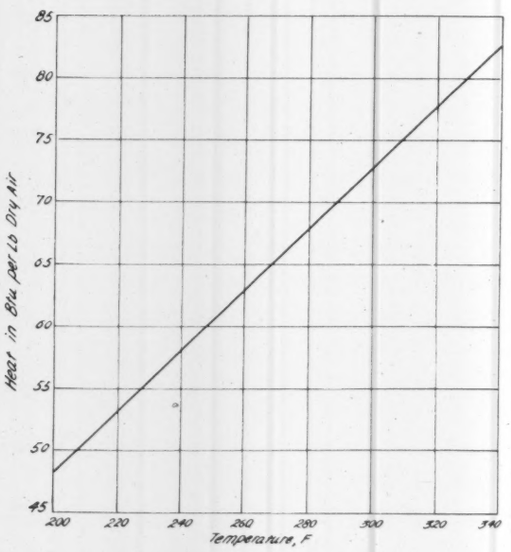


FIG. 7—HEAT CONTENT OF DRY AIR

ture to be removed it is evident that the heat used for evaporation is 1,250,800 Btu. As the temperature drops from 230 to 190 the heat expended on vaporization is equal to 40 x 0.26 or 10.4 Btu per lb. of dry air. When the total heat is divided by this amount it is learned that 12,027 lb. of dry air are moved in the second stage. Dividing the total moisture to be removed by this quantity gives a humidity of 0.109 for the air entering the third stage.

Third Stage: In the third stage the humid heat is read from the chart as 0.289 Btu and the heat to evaporate 1 lb. of water is 981½ Btu. Multiplying the latter amount by the weight of moisture to be removed, or 525 lb., we obtain 514,690 Btu provided for evaporation of moisture. Assuming that the temperature drop is 70 F in this last stage the heat for vaporization of 1 lb. of dry air is 70 x 0.289 or 20.23 Btu. Dividing the total heat by this amount gives 25,442 lb. of dry air. As it is desired to avoid adding additional heat in either of the last two stages it will be necessary that some of the air from the first stage be by-passed to the third stage.

Charts for High-Temperature Drying

The problem of high-temperature drying is not so easily solved by psychrometric charts. The reason is that, for temperatures above 212 F, the partial pressures for the water vapor alone will exceed 29.92 in. of mercury which is the total air pressure used in the design of such charts. To meet this need four charts, Figs. 4 to 7, have been prepared. Fig. 4 gives the moisture in air-water vapor mixtures where temperatures are high and the partial pressures of the air alone range from ½ to 5 lb. Fig. 5 gives the volumes for saturated mixtures under similar conditions. Fig. 6 is identical with Fig. 5 except that the mixtures are absolutely dry. Fig. 7 gives the heat content of bone-dry air; with it and a steam table, which will furnish the heat content of the vapor when saturated, the total heat per lb. of dry air can be calculated.

The weight of moisture in air-water vapor mixtures at high temperatures can be calculated through the following thermodynamic relation:

$$V_m P_w = W_w R_w T_m$$

where V_m is the volume of the mixture
P_w is the water vapor pressure
W_w is the weight of water vapor
R_w is the gas constant for water vapor
T_m is the absolute temperature of the mixture

One of the biggest problems that we are facing is the smoke nuisance. Smoke is injurious to health. It is expensive in that it means fuel waste, high laundry bills, defacement of expensive buildings and lessened working capacity. Experts declare that throughout the United States smoke costs each inhabitant $16 annually.

The remedy for the industrial smoke problem is to provide such engineering oversight of new installations as to insure a non-smoking plant to begin with. It may mean changes in building plans, a larger investment, a greater responsibility of management, a more intelligent operation, but these things are the price of clean air. — E. R. Weidlein, *Journal* of the *Western Society of Engineers.*

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

1895

APRIL 1932

Material in this Section prepared by Publication Committee: F. D. Mensing, *Chairman;* E. O. Eastwood and F. E. Giesecke
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Performance of Convector Heaters

By A. P. Kratz[1] (*MEMBER*) and M. K. Fahnestock[2] (*NON-MEMBER*)
Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory

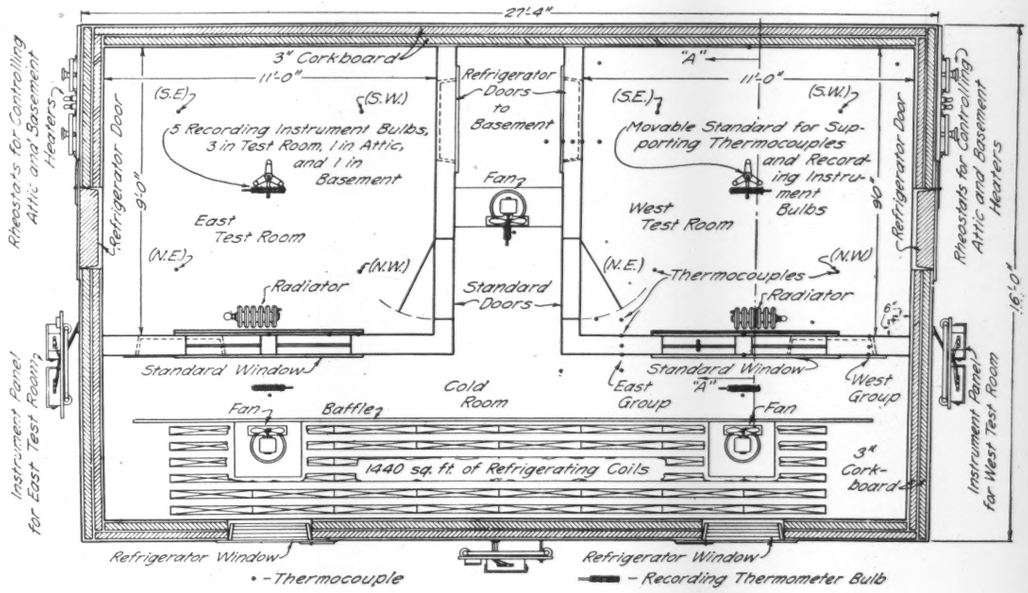


FIG. 1—PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work is carried on in the Department of Mechanical Engineering under the general supervision of A. C. Willard, Professor of Heating and Ventilation and head of the Department. Acknowledgment is due E. L. Broderick, Research Assistant in Mechanical Engineering, for valuable assistance in the collection and preparation of the data.

THE object of this part of the investigation was to make a comparison of the performance of four types of cabinet convector heaters with the performance of an unenclosed direct cast-iron radiator of approximately the same height when a temperature of 68 F was maintained at a level 30 in. from the floor in all cases. A further object was to compare the performance of the convector heaters with that of a tubular cast-iron radiator enclosed in a cabinet conforming to the space requirements of the convector heaters, and operated under the same conditions.

[1] Research Professor in Mechanical Engineering, University of Illinois.
[2] Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
[3] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223. Also TRANSACTIONS of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 35, 1929, pages 79-89.

Description of Apparatus

All tests were conducted in the low temperature testing plant which has been completely described in previous publications.[3] In brief, this plant, as shown in Figs. 1 and 2, consists of two rooms erected within a larger room having 6-in. cork walls, floor and ceiling. The latter room is equipped with direct expansion refrigerating coils, and each of the two test rooms presents two walls of standard frame construction exposed to any desired temperature maintained in the cold room. One wall of each test room contains a pair of double hung windows and the other wall a standard door with a glass panel. In addition, arrangements are made so that the floors and ceilings of the test rooms may be exposed to any desired temperatures. Air movement over the walls is obtained by means of fans.

The plant is thoroughly equipped with thermocouples and temperature recorders in order to permit observations being made without the necessity for entering the rooms and thus disturbing conditions at any time during the period of a test. The radiators are connected with a single pipe for steam and return condensation, and the condensation is weighed as shown in Fig. 2. A steam separator is installed in the line to each radiator, and an electrical superheater further insures the use of dry steam. Provision is also made for venting the radiators to prevent the accumulation of air.

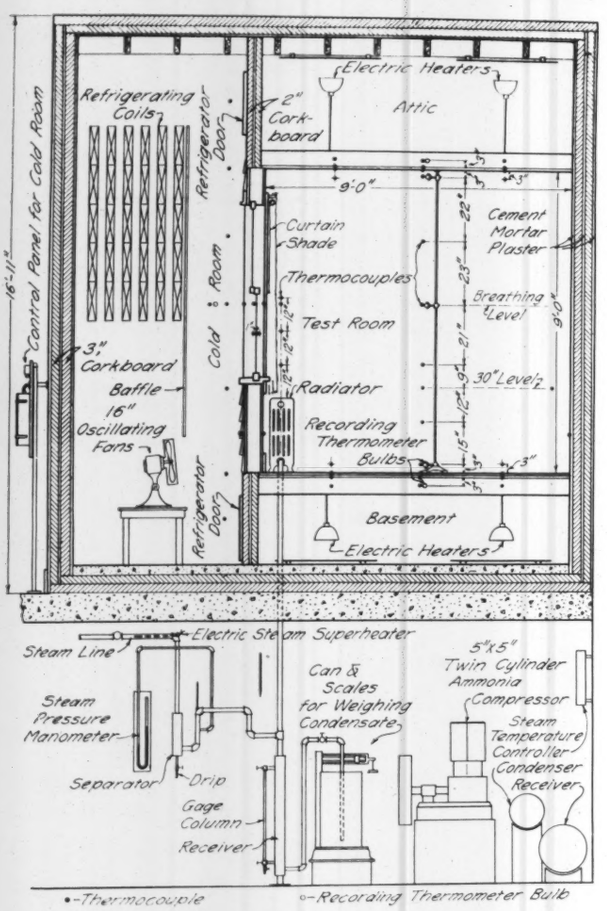
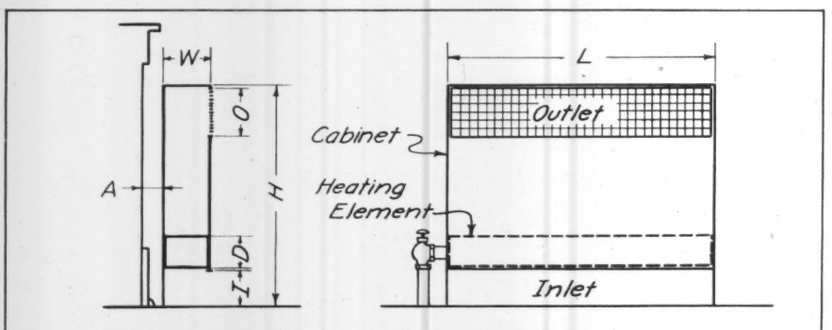


FIG. 2—ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT



Convector Heater No.	Type of Heating Element	Dimensions in Inches						
		A	W	H	L	I	O	D
1	Cast Iron	2½	5½	26	36'	4	5	14 7/16
2	Non-ferrous	2½	5	30 3/8	31	6	6	6
6	Non-ferrous	2½	5½	28	30[2]	4	5	2
7	Bimetallic	2½	6 1/16	28 3/8	22 5/8[2]	4 1/8	3 1/8	20
Enclosure No.								
24	3-Tube C.I. Rad.	2	5 3/4	27	31½	4¼	6[3]	26
24d	3-Tube C.I. Rad.	2	5 3/4	27	34	4¼	6	26
24e	3-Tube C.I. Rad.	2	5 3/4	27	34[4]	4¼	22¼	26

2-18-32

[1] "6" of length blocked. [2] "3" of length blocked.
[3] Open outlet.
[4] 1-Section Wrapped.

FIG. 3—DIMENSIONS OF CONVECTOR HEATERS AND ENCLOSED CAST-IRON RADIATOR TESTED

The temperatures at the various levels within the test rooms are observed by means of thermocouples made from No. 22 B and S gage copper and constantan wire and supported on a standard in the center of the room as shown in Fig. 2. A study of temperature conditions within the test rooms proved that the center reading at a given level is representative of the average temperature at that level.

Four representative convector heaters of the cabinet type were used. The dimensions of the cabinets are shown in Fig. 3, and further details of the heating elements are shown in the insets in Figs. 4 to 7. Convector heater No. 1 contained a cast-iron heating element with cast-iron fins. Heaters Nos. 2 and 6 contained non-ferrous heating elements with thin metal fins. Heater No. 7 contained a non-ferrous heating element with thin metal fins, combined with a cast-iron front, the latter partly filled with steam and serving as direct panel radiation.

In addition, tests were run on a 13-section, 26-in., 3-tube cast-iron direct steam radiator enclosed in a cabinet similar in design and space requirements to those used in connection with the convector heaters. Three types of cabinets were used with this radiator, as shown in Fig. 3, and in the inset in Fig. 8. They consisted of (1) Enclosure No. 24, a cabinet with open inlet and outlet, (2) Enclosure No. 24d, with an open inlet and grilled outlet, and (3) Enclosure No. 24e, with a fully grilled front extending to within 4¼ in. of the floor. An 8-section, 26-in., 5-tube unenclosed direct cast-iron radiator was also tested, and was used as a standard for comparison in all cases.

Test Procedure

In all cases, the temperature in the cold room was maintained at about —2.0 F and one of the exposed walls of the test room was subjected to an equivalent wind velocity of approximately 10 mph. The temperature of the air above the ceiling of the test room was maintained at 62 F and the air in the basement at such a temperature that the upper surface of the floor was approximately two degrees warmer than the lower surface. No test observations were made until conditions had remained constant for several hours, as indicated by the readings of the thermocouples on the inside wall surfaces. This required about 20 hours after a change in radiators was made. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min. increments of weight. At the end of each test a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

The output of the heating unit was adjusted, by wrapping part of the sec-

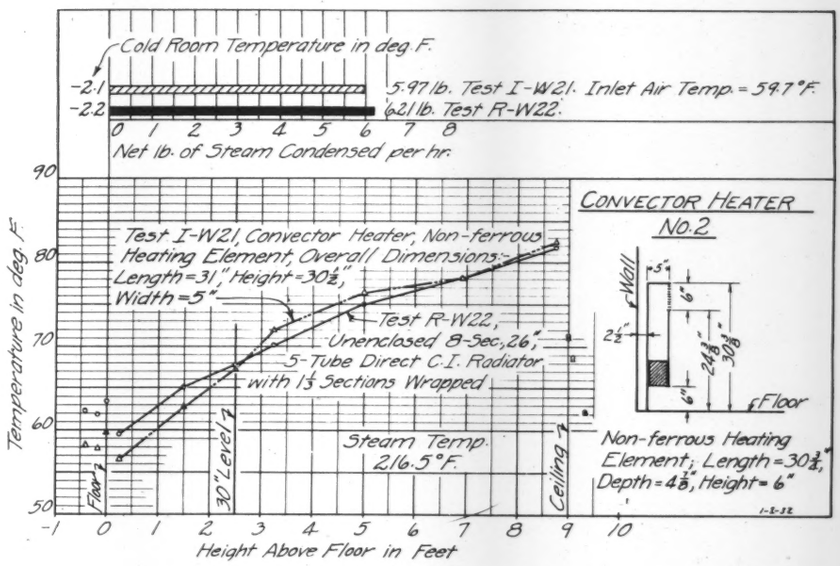

Cold Room Temperature in deg. F.
-2.1
-2.2
5.97 lb. Test I-W21. Inlet Air Temp. = 59.7°F.
6.21 lb. Test R-W22.
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
CONVECTOR HEATER No. 2
Test I-W21, Convector Heater, Non-ferrous Heating Element, Overall Dimensions:- Length = 31", Height = 30½", Width = 5"
Test R-W22, Unenclosed 8-Sec., 26", 5-Tube Direct C.I. Radiator with 1⅓ Sections Wrapped
Steam Temp. 216.5°F.
Wall
Floor
Ceiling
30" Level
Non-ferrous Heating Element; Length = 30½", Depth = 4⅛", Height = 6"
Temperature in deg. F.
Height Above Floor in Feet

FIG. 4—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A NON-FERROUS HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

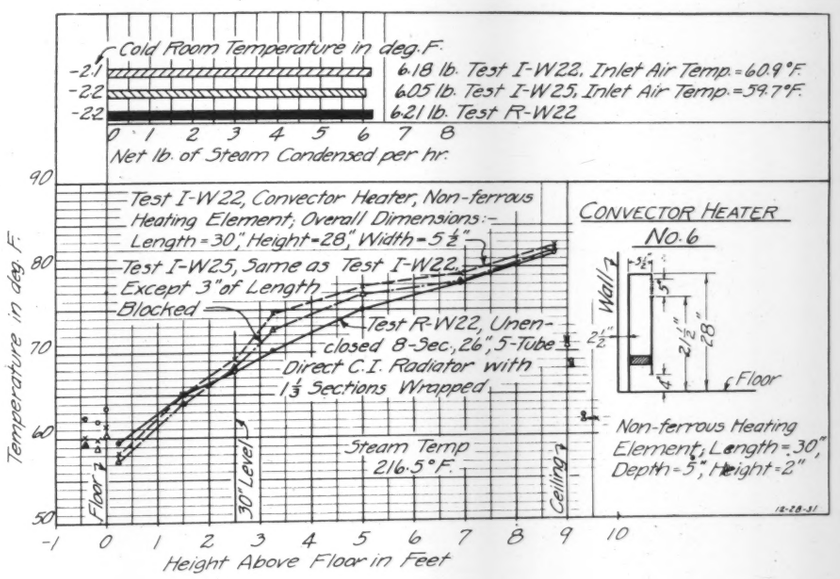

Cold Room Temperature in deg. F.
-2.1
-2.2
-2.2
6.18 lb. Test I-W22. Inlet Air Temp. = 60.9°F.
6.05 lb. Test I-W25. Inlet Air Temp. = 59.7°F.
6.21 lb. Test R-W22
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
CONVECTOR HEATER No. 6
Test I-W22, Convector Heater, Non-ferrous Heating Element, Overall Dimensions:- Length = 30", Height = 28", Width = 5½"
Test I-W25, Same as Test I-W22, Except 3" of Length Blocked
Test R-W22, Unenclosed 8-Sec., 26", 5-Tube Direct C.I. Radiator with 1⅓ Sections Wrapped
Steam Temp. 216.5°F.
Wall
Floor
Ceiling
30" Level
Non-ferrous Heating Element; Length = 30", Depth = 5", Height = 2"
Temperature in deg. F.
Height Above Floor in Feet

FIG. 5—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A NON-FERROUS HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

tions of the cast-iron radiators or by blocking off part of the cabinets and heating elements of the convector heaters, so that each unit maintained a temperature of 68 F at a level 30 in. above the floor under the standard conditions of exposure previously outlined.

Results of Tests

The results of the tests on the convector heaters are shown in Figs. 4 to 7 inclusive. The full line curve in every case represents the performance of the unenclosed 8-section, 26-in., 5-tube cast-iron radiator with 1 1/3 sections wrapped, which was used as the standard for comparison. In Figs. 5, 6, and 7 the temperature gradient curves for convector heaters Nos. 6, 7, and 1 respectively without any blocking are shown. These heaters were chosen from the estimated heating requirements and the stock sizes furnished by the manufacturers. In each case the nearest stock size slightly over-heated the room, and it was necessary to block off part of the cabinet and heating element in order to obtain a temperature of 68 F at the 30-inch level in the test room. The fact that the temperature gradient curves for the blocked heaters were practically parallel to those for the unblocked heaters is evidence that the small amount of blocking necessary had no influence on the characteristic performance insofar as the heating effect was concerned. The comparisons have, therefore, been based on the steam condensation and temperature gradient curves for the blocked heaters.

From Figs. 4 to 6 inclusive, it may be noted that convector heaters Nos. 2, 6, and 7 showed practically the same characteristics. The convector heater in each case gave somewhat lower temperatures than the unenclosed cast-iron radiator in the zone from the floor to the 30-in. level, higher temperatures in the zone between the 30-in. level and the 7-ft level, and either the same or slightly higher temperatures in the zone above the 7-ft level.

Convector heaters Nos. 2, 6, and 7 maintained a temperature of 68 F at the 30-in. level with reductions in steam condensation of 4.0, 2.5, and 1.5 per cent respectively, as compared with the steam condensation of the unenclosed cast-iron radiator under the same conditions.

Convector heater No. 1 exhibited the same general characteristics as the others except that the zone of higher temperature extended only as high as the 4.5-ft level and that the temperatures in the zone above this level were lower than those given by the unenclosed cast-iron radiator. Hence, while the resulting comfort conditions were not essentially different from those produced by the other convector heaters, a slight gain in economy was effected by the reduction in the temperatures above the 4.5-ft level. In this case, the reduction in steam condensation was 7.1 per cent as compared with that for the unenclosed cast-iron radiator.

The performance curves for the 26-in., 3-tube, cast-iron radiator enclosed in three types of cabinets are shown in Fig. 8. These three cabinets differed only in the character of the air inlets and outlets, and the overall dimensions corresponded approximately to those for the cabinets of the convector heaters. It may be noted from Fig. 8 that the performance of the enclosed radiator is compared directly with the performance of the same unenclosed 26-in., 5-tube, cast-iron radiator used in comparison with the convector heaters, and that the enclosed cast-iron radiator exhibited practically the same characteristics as the convector heaters. While some slight adjustments in the size of the radiator with the different types of air inlets and outlets were required,

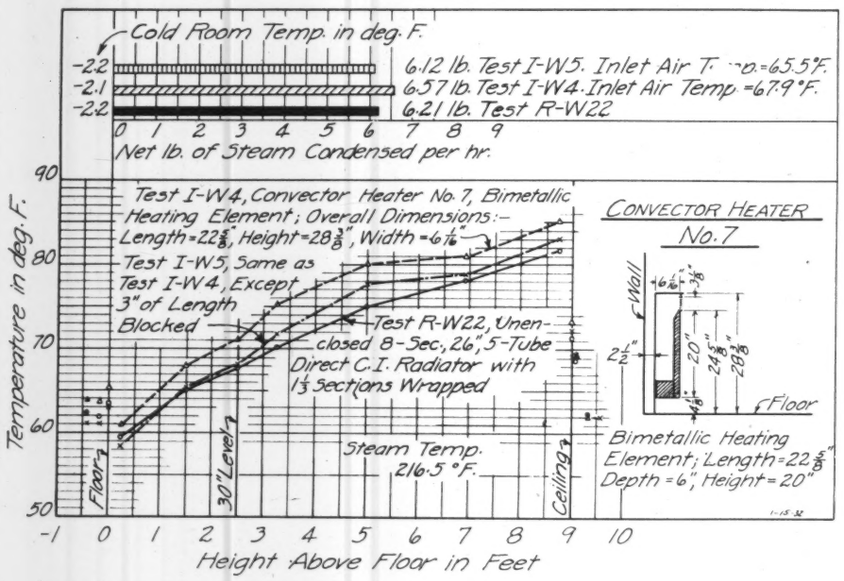

Cold Room Temp. in deg. F.
-2.2
-2.1
-2.2
6.12 lb. Test I-W5. Inlet Air T. ·· ·0.=65.5°F.
6.57 lb. Test I-W4. Inlet Air Temp. =67.9°F.
6.21 lb. Test R-W22
0 1 2 3 4 5 6 7 8 9
Net lb. of Steam Condensed per hr.
Test I-W4, Convector Heater No. 7, Bimetallic Heating Element; Overall Dimensions:- Length=22⅞", Height=28⅜", Width=6⅛"
Test I-W5, Same as Test I-W4, Except 3" of Length Blocked
Test R-W22, Unenclosed 8-Sec., 26", 5-Tube Direct C.I. Radiator with 1½ Sections Wrapped
CONVECTOR HEATER No. 7
Wall
Floor
Steam Temp. 216.5 °F.
Floor
30" Level
Ceiling
Bimetallic Heating Element; Length=22⅝" Depth=6", Height=20"
Temperature in deg. F.
90
80
70
60
50
-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG. 6—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A BIMETALLIC HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

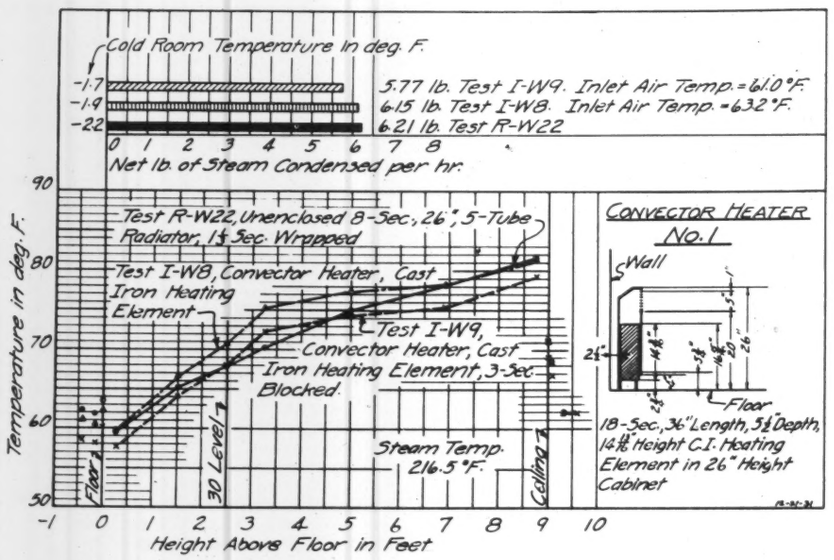

Cold Room Temperature in deg. F.
-1.7
-1.9
-22
5.77 lb. Test I-W9. Inlet Air Temp.=61.0°F.
6.15 lb. Test I-W8. Inlet Air Temp.=63.2°F.
6.21 lb. Test R-W22
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
Test R-W22, Unenclosed 8-Sec., 26", 5-Tube Radiator, 1½ Sec. Wrapped
Test I-W8, Convector Heater, Cast Iron Heating Element
Test I-W9, Convector Heater, Cast Iron Heating Element, 3-Sec. Blocked
CONVECTOR HEATER No. 1
Wall
Floor
Steam Temp. 216.5 °F.
Floor
30" Level
Ceiling
18-Sec., 36" Length, 5½" Depth, 14⅛" Height C.I. Heating Element in 26" Height Cabinet
Temperature in deg. F.
90
80
70
60
50
-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG. 7—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A C. I. HEATING ELEMENT WITH THE SAME AIR TEMPERATURE AT THE 30-IN. LEVEL

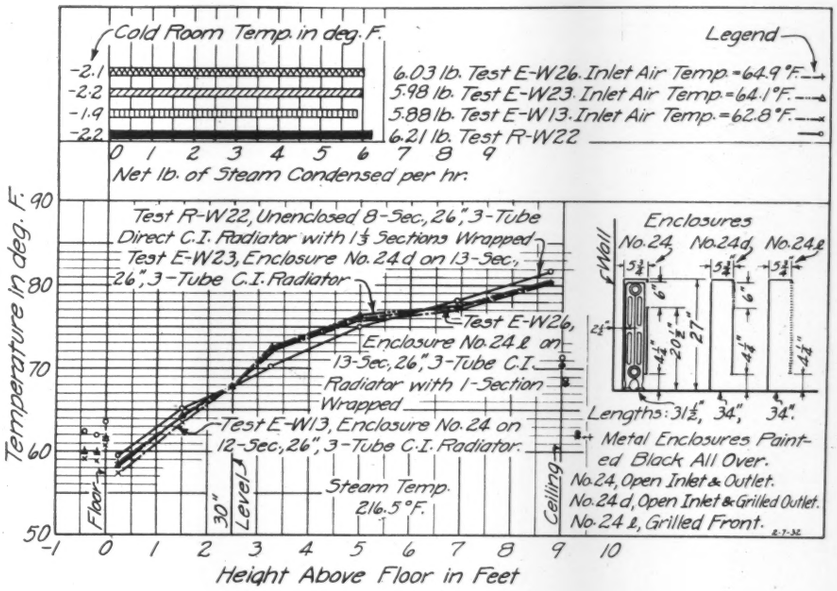

Cold Room Temp. in deg. F.
-2.1
-2.2
-1.9
-2.2
Legend
6.03 lb. Test E-W26. Inlet Air Temp. = 64.9 °F.
5.98 lb. Test E-W23. Inlet Air Temp. = 64.1 °F.
5.88 lb. Test E-W13. Inlet Air Temp. = 62.8 °F.
6.21 lb. Test R-W22
0 1 2 3 4 5 6 7 8 9
Net lb. of Steam Condensed per hr.
Temperature in deg. F.
90
80
70
60
50
Test R-W22, Unenclosed 8-Sec., 26", 3-Tube Direct C.I. Radiator with 1½ Sections Wrapped
Test E-W23, Enclosure No. 24 d on 13-Sec., 26", 3-Tube C.I. Radiator
Test E-W26, Enclosure No. 24 e on 13-Sec, 26", 3-Tube C.I. Radiator with 1-Section Wrapped
Test E-W13, Enclosure No. 24 on 12-Sec, 26", 3-Tube C.I. Radiator
Steam Temp. 216.5 °F.
Floor
30" Level
Ceiling
Enclosures
No. 24
No. 24 d
No. 24 e
Wall
Lengths: 31½", 34", 34".
Metal Enclosures Painted Black All Over.
No. 24, Open Inlet & Outlet.
No. 24 d, Open Inlet & Grilled Outlet.
No. 24 e, Grilled Front.
-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG. 8—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED 5-TUBE DIRECT C. I. RADIATOR AND AN ENCLOSED 3-TUBE DIRECT C. I. RADIATOR

the temperature gradient curves and steam condensations were all practically the same when 68 F was maintained at the 30-in. level. The steam condensation, as compared with that for the unenclosed radiator, was reduced 5.5, 3.8, and 3.0 per cent by Enclosures Nos. 24, 24d, and 24e, respectively. While Enclosure No. 24, with the open inlet and outlet, gave slightly less steam condensation than those with grilled outlets, it also produced somewhat lower temperatures at the floor. On the whole, the performance of the enclosed cast-iron radiator was not essentially different from that of the convector heaters.

Conclusions

From the results of these tests, the following conclusions may be drawn:

1. The four types of convector heaters, having cabinets of approximately the same height, did not differ essentially in performance characteristics.

2. The convector heaters produced temperatures slightly lower in the zone from the floor to the 30-in. level, higher in the zone from the 30-in. level to the 7-ft level, and either the same or slightly lower in the zone above the 7-ft level, than those obtained with the unenclosed 26-in., 5-tube cast-iron radiator.

3. The convector heaters maintained a temperature of 68 F at the 30-in. level with steam condensations ranging from 1.5 to 7.1 per cent less than the steam condensation required by the unenclosed 26-in., 5-tube cast-iron radiator.

4. A 26-in., 3-tube, cast-iron radiator enclosed in a cabinet of similar type and conforming approximately with the space requirements of the cabinets used with the convector heaters, exhibited the same performance characteristics as the convector heaters.

Leakage of Gases Into Cellar in Operation of Domestic Heating Furnaces

In the operation of all domestic heating furnaces, it is very necessary to adjust the dampers so that gases from the burning fuel are not forced into the cellar or, in warm-air furnaces, into the air ducts, and thus to the rooms of the house, cautions the United States Bureau of Mines. These gases are objectionable not only because of smoke and dirt, and because the sulphur in them tarnishes silver-ware, but also because they may contain carbon monoxide; the presence of carbon monoxide is more probable when there is pressure in the furnace, because under such conditions, no air to burn the combustible gases arising from the fuel bed is drawn into the furnace through the slots in the firing door. As long as a draft is maintained in the combustion space over the fuel, all the gases will pass to the stack, but if the pressure in that space is greater than that of the cellar, some of the gases will be forced into the cellar or, if there are leaks from the fire-pot to the air ducts, into the air system.

If there is a pressure in the furnace it is indicated by smoke or fumes coming out of the firing door or one of the upper clean-out doors or, if the heater is not in good repair, through the cracks. There are two methods by which the emission of fumes can be stopped. The more common method is to increase the draft by adjusting one of the stack dampers, but the same end can be accomplished by restricting the flow of air into the ash pit by partly closing its damper. If the first method is used it will result in a somewhat increased rate of burning. A pressure, instead of a draft in the furnace, is more likely to occur when the fuel bed does not offer much resistance to the passage of the air from the ash pit. This resistance depends on the size of the fuel and is greater as the fuel is smaller, or as a coal cakes and cokes and closes up the openings. For free-burning coals and for cokes one depends on the resistance of the layer of ash at the bottom of the bed to make up for the lack of resistance of the fuel itself. When burning coke, it is very desirable to maintain a layer of ashes as the detection of the emission of gases is more difficult since there is no smoke which can act as an indicator.

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Heat Transmission as Influenced by Heat Capacity and Solar Radiation

By F. C. Houghten[1] (*MEMBER*), J. L. Blackshaw[2] (*MEMBER*), E. M. Pugh[3] (*NON-MEMBER*) and Paul McDermott[4] (*NON-MEMBER*), Pittsburgh, Pa.

THE fact that solar radiation is an important factor in the mechanism of heat flow into and out of buildings has long been recognized by heating and ventilating engineers. It is usual, however, in practical heat transmission problems to neglect the sun effect entirely, to employ coefficients determined under steady state conditions, and to consider the average inside and outside temperatures constant throughout the day. With this practice, factors are used to take care of wind, sun effect or its absence, or other unusual conditions in order to insure the selection of a heating plant sufficiently large to handle the load on the coldest day, or an air-conditioning system of sufficient capacity for the warmest weather.

Research, carried on by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Pittsburgh Station of the United States Bureau of Mines, has shown that a large error[5] may be introduced into the calculations by failure to consider the periodic character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the walls into a building on a hot, sunny day.

Scope of the Present Work

This paper reports the characteristics of heat flow through eight roof panels which were studied at the Laboratory during the summer months of 1930 and 1931. It gives the cyclic heat flow through the inner surfaces of these panels, the maximum and minimum rates of penetration during twenty-four hour periods, the timing and intensity of the temperature waves at the outer and inner surfaces, and the rate of heat flow into an air-conditioned space, as these are determined by the thickness, conductivity and heat capacity of the different panels studied.

A theoretical mathematical analysis of the cyclic flow of heat under the foregoing conditions results in the presentation of formulae which give the rate of heat flow into an air-conditioned space for any hour of the day as a function of a Fourier series analysis of the outside surface temperature, the air-conditioned temperature, and the thickness and the physical properties

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of the structure. An empirical method of determining certain constants in these theoretical formulae is developed. This simplified method gives the heat flow into an air-conditioned space with a fair degree of accuracy, in terms of the maximum and minimum outside surface temperatures for the day, the inside air-conditioned temperature, and the thickness and physical properties of the structure.

The paper also includes data on the conductivities, densities and specific heats of the materials contained in the different roof panels, and the methods developed for obtaining them. A recording pyrheliometer, developed at the A. S. H. V. E. Laboratory for measuring solar radiation, is described, and data are presented for the first days of July, August and September, on the intensity of solar radiation in Pittsburgh, and of the resulting intensity of solar radiation impingement upon a horizontal surface, and upon east, south and west walls.

Factors Affecting Periodic Heat Flow

The flow of heat into an air-conditioned space through a roof or wall on a hot summer day, because of the higher outside air temperature and the solar radiation on the outer surface, is very complex. It may be illustrated

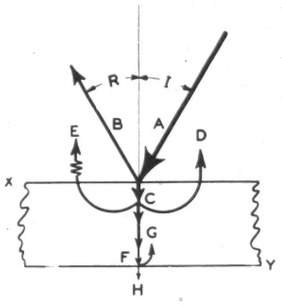


FIG. 1 — FACTORS AFFECTING HEAT FLOW THROUGH A ROOF

in Fig. 1, where XY is the roof or wall structure. The impingement of solar radiation against the outside surface is represented by A. If it were not for the earth's atmosphere with its clouds, haze and dust, the intensity of the solar radiation in Btu per square foot per hour perpendicular to the direction of the sun's rays would be practically constant, showing only a small seasonal variation caused by the change in the sun's distance from the earth. The intensity of solar radiation is decreased by absorption by the earth's atmosphere, especially in the morning and evening, when the sun must shine through a much greater thickness of the atmosphere in reaching the earth's surface. The Weather Bureau has reported 338 Btu per hour per square foot normal to the direction of the sun's rays for a clear day at Washington. The Laboratory has actually measured 311 Btu at noon of a clear day in Pittsburgh.

The intensity of impingement on the surface, X, not normal to the sun's rays, decreases with the angle of incidence, I, which is a function of time; a part of the energy, B, is not absorbed, but is reflected away. The magnitude of B is a function of the character of the surface, X, and of the angle of reflection, R, which varies with the time of day. The remaining energy, C, is absorbed as heat by the surface, X, raises its temperature enough to cause the radiation, D, back through the air and the convection loss, E, by direct contact with the air. Also, as the temperature of the surface, X, is raised by the absorption of radiant energy, heat flows in the direction, F. As the temperature of the surface, X, rises to a maximum at noon and then recedes, a wave of heat advances towards F. Because heat is required to raise the temperature of each increment of the distance, X to Y, the rate of heat flow past any point between X and Y diminishes as the wave penetrates through the structure, and when the wave reaches the surface, Y, it has a much lowered amplitude, dependent upon the conductivity, density, specific heat and thickness of the material in the wall, the film resistance of its lower surface, and the temperature of the air below. The crest of the wave reaching the surface, Y, will be delayed a certain time after the crest at the surface, X.

Conductivity, density and specific heat are factors which combine to damp out the wave amplitude. In a theoretical consideration of heat transfer they are combined into a single constant (h), called the diffusivity:

$$h = \sqrt{\frac{k}{c\rho}} \qquad (1)$$

where

h = diffusivity
k = conductivity
c = specific heat
ρ = density

This combined constant takes into account the resistance to heat flow and the heat capacity of the structure.

The heat reaching the lower surface tends to build up its temperature, and a small amount of heat, G, returns in a reflected wave upward towards X. The magnitude of this reflected wave also depends upon the physical properties of the structure. Only H, the heat entering the air-conditioned space below, is of interest to the air-conditioning engineer.

An ideal solution of the problem of heat flow as affected by solar radiation would give the value of H for all hours of the day as a function of the intensity of solar radiation perpendicular to the direction of the sun's rays, of the atmospheric conditions, and of the physical characteristics of the structure. Such a solution involves the following factors: the intensity of solar radiation, A, perpendicular to the direction of the rays; the angle of incidence, I, of impingement against the surface; the reflected energy, B, depending upon the angle of reflection, R; the radiated heat, D, depending upon the convection from the surface, X; the difference in temperature of the surface, X, and the outside air; the temperature difference between the surfaces, X and Y; the conductivity, the density, specific heat and thickness of structure; the film resistance for the surfaces, X and Y; and the controlled temperature of the air below. This makes thirteen factors upon which H depends, the first five of which are harmonic functions of time.

The problem of determining the value of H may be conveniently divided into two distinct parts, each of which is complicated: *first*, a determination of the varying temperatures of the surface, X, depending upon the values of A, B, C, D, E, and G, and the physical properties of the structure; and, *second*, a determination of the value of H at all times during the daily cycle, which value depends upon the change of the temperature of the surface, X, the physical properties of the structure.

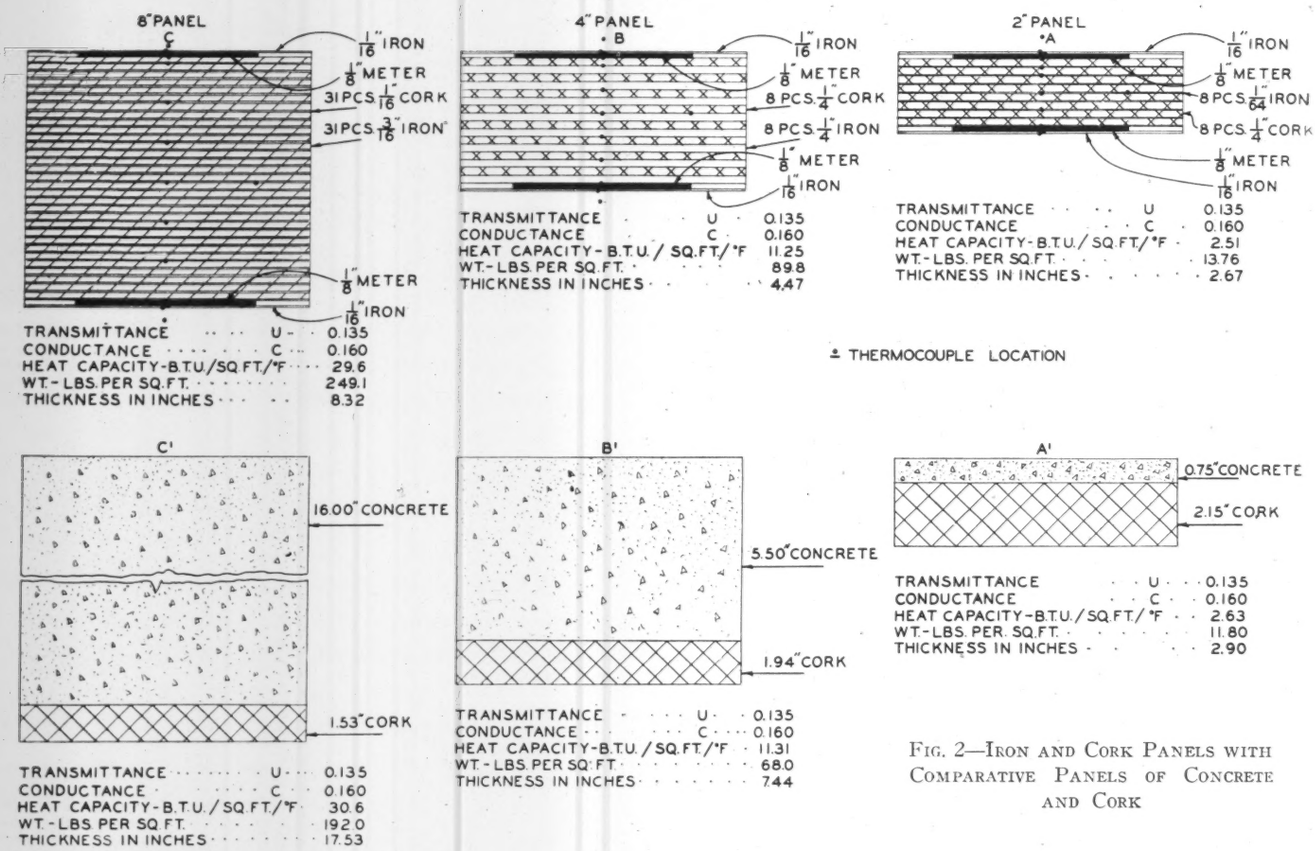


FIG. 2—IRON AND CORK PANELS WITH COMPARATIVE PANELS OF CONCRETE AND CORK

the film resistance of the inner surface, and the constant air temperature in the space below. Only the second of these two parts of the entire problem is treated in this paper, namely, the evaluation of H from the cyclic change in the temperature of the surface, X, and the characteristics of the structure XY.

Roof Panels Studied

At the beginning of the investigation it was considered desirable to study characteristics of heat penetration for three panels having approximately the same conductances but widely different heat capacities. Since it was desirable to eliminate as many variables as possible, and at the same time to have panels which could be considered homogeneous to the extent that the material in any small cube in the panel would be exactly similar to that of any other similar cube in the same panel, roof panels were built up by laminating alternate sheets of iron and cork, which afforded an opportunity to distribute thermocouples through the layers. Cross-sections of these panels, studied in the summer of 1930, are shown in Fig. 2. For convenience in comparing the iron and cork panels with structures familiar to the heating and ventilating engineer, the figure also shows panels of concrete and cork having approximately the same conductance and heat capacity.

During the summer of 1931, a 6-in. concrete, a 4-in. gypsum, and a 2-in. yellow pine panel were studied, as was also a panel made up of 3-in. concrete and 1-in. cork, tested first with concrete upwards, and later with the cork on top. (In the following references to this panel, the first term of the compound word, cork-concrete or concrete-cork, indicates the side of the panel which was upward during test.) All panels were 3 ft square. The concrete used was 4:2:1 mix of gravel, sand and cement with a 4½-in. slump. The concrete was poured onto smooth 1-in. corkboard at the bottom of the form so that concrete and cork were tightly bonded together and remained so throughout the tests. The gypsum panel was made of commercial roofing gypsum mixed according to specifications for a gypsum roof supplied by a large manufacturer. The concrete, the cork-concrete, and the gypsum panels were poured by Charles W. Larkin, Head Instructor of Masonry Construction at Carnegie Institute of Technology. These panels were perfect as regards uniformity of thickness and smoothness of surfaces. The yellow pine panel was made by gluing together 2¼-in. x 6-in. strips of yellow pine and dressing them down to a panel accurately 2 in. in thickness with smooth planed surfaces.

The over-all physical properties of the seven panels are given in Table 1. The over-all conductivity values

TABLE 1—PHYSICAL PROPERTIES OF PANELS

DESCRIPTION OF PANEL	THICKNESS l		CONDUCTIVITY k		DENSITY ρ		SPECIFIC HEAT c	INNER FILM CONDUCTANCE, h_i	
	ENG.	METRIC	ENG.	METRIC $\times 10^{-4}$	ENG.	METRIC		ENG.	METRIC $\times 10^{-4}$
2" Laminated Iron & Cork	2.67	6.78	.427	1.470	61.68	.988	.192	1.0	1.355
4" Laminated Iron & Cork	4.47	11.35	.734	2.528	238.77	3.826	.128	1.25	1.694
8" Laminated Iron & Cork	8.32	21.13	1.231	4.240	357.05	5.722	.119	1.25	1.694
2" Pine	2.156	5.476	.830	2.855	37.46	.600	.467	1.9	2.574
6" Concrete	6.188	15.717	8.00	27.520	148.6	2.381	.230	1.9	2.574
4" Gypsum	4.188	10.637	1.445	4.971	64.89	1.040	.234	1.9	2.574
4" Cork-Concrete & 4" Concrete-Cork	4.188	10.637	1.218	4.190	101.03	1.619	.215	1.9	2.574

All values based on average between outer and inner surfaces.

were obtained either directly by testing the panels as placed in their set-up, or by averaging the known conductivities of their homogeneous component parts. The over-all densities and specific heats are averages for the entire individual panel as constructed. The densities and specific heats of the component homogeneous parts were determined by test, except in a few cases where they are handbook data. The physical properties of the homogeneous materials going into the various panels are given in Table 2.

TABLE 2—PHYSICAL PROPERTIES OF MATERIALS USED IN PANELS

MATERIAL	CONDUCTIVITY k	DENSITY ρ	SPECIFIC HEAT, c
Iron Plate	33.3	489.5	.116
Gasket Cork	.35	14.43	.43
Pine	.89	37.45	.47
Concrete	15.0	148.6	.230
Gypsum	1.58	64.9	.234
Paper & Heat Meter	5.51(a)	45.1	.35
Corkboard	.35	15.90	.43

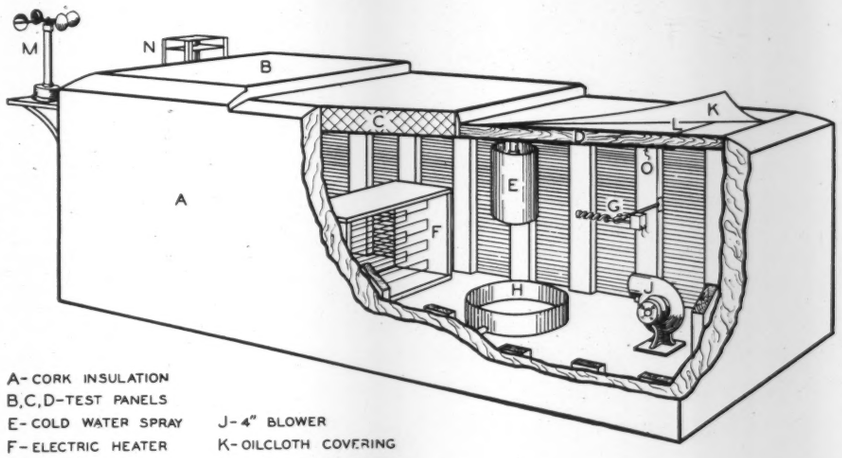
Test Set-up

The test equipment for studying the effect of solar radiation on heat transfer was built on the flat roof of a low building at the Pittsburgh Station of the United States Bureau of Mines, so situated that at no time during the day was the roof without an unrestricted view of the sun. This set-up provided for the simultaneous study of three panels, each 3 ft. square, which formed a roof over an air space 3 ft. high by 3 ft. wide by 9 ft. long as shown in Fig. 3 and Fig. 4. The floor and side walls of this chamber were insulated with 4 in. of corkboard, *A*, tarred on the outside to prevent air infiltration and to furnish a weatherproof covering. The three test panels, *B, C, D,* form the roof of the air-conditioned space, and are so placed that their lower surfaces are in the same plane. The cork insulation on the sides of the air-conditioned space extended to the upper surfaces of the three panels so as to insulate the edges of the panels, and the joints between the panels and the insulation, and the joints between the edges of different panels were sealed with tar.

Provision for maintaining constant temperatures and humidities in the space below the roof panels was furnished by an ice-cold water spray, *E*, the well-distributed low-intensity 500 watt electric heater, *F*, and the thermostat, *G*. Cold water was pumped through insulated piping to the cooling spray by a rotary gear pump from an insulated ice bath in the room below. The spray water was caught in the receiver, *H*, and returned to the ice bath by gravity. By varying the volume of spray water, the amount of cooling needed to maintain the air space at any predetermined temperature was readily controlled. The spray maintained the relative humidity of the conditioned air a constant, sufficiently low to prevent condensation on any of the interior surfaces. When heating was necessary, the thermostat, *G*, by regulation relays, controlled the current passing through the heater, *F*.

The thermostat held temperatures sufficiently constant for periods preliminary to the actual tests; for the accurate conditions necessary during tests, the heat was controlled manually by either graduating the current through the heater, *F*, or by adjusting the volume of cold water passing through the spray, using an orifice meter and hand-controlled valves. To prevent stratification of the air, a small slow speed blower, *J*, was located on the floor at one end of the chamber. The air inlet of this blower was restricted by a damper to allow only a small volume of air to be moved at low velocity. This agitated the air in the box enough to prevent

FIG. 3—TEST CHAMBER WITH THREE ROOF PANELS UNDER TEST



A-CORK INSULATION
B,C,D-TEST PANELS
E-COLD WATER SPRAY
F-ELECTRIC HEATER
G-THERMOSTAT
H-SPRAY RECEIVER
J-4" BLOWER
K-OILCLOTH COVERING
L-HEAT FLOW METER
M-CUP ANEMOMETER
N-SUN SHIELD FOR AIR COUPLE

FIG. 4—AIR CONDITIONED CHAMBER FOR ANALYZING HEAT FLOW THROUGH THREE ROOF PANELS

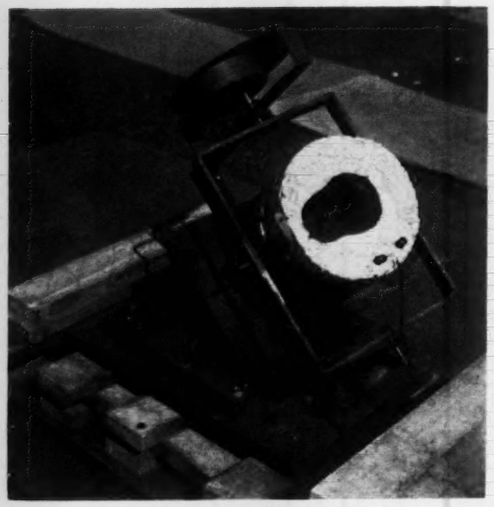


FIG. 5—PYRHELIOMETER FOR AUTOMATICALLY RECORDING INTENSITY OF SOLAR RADIATION

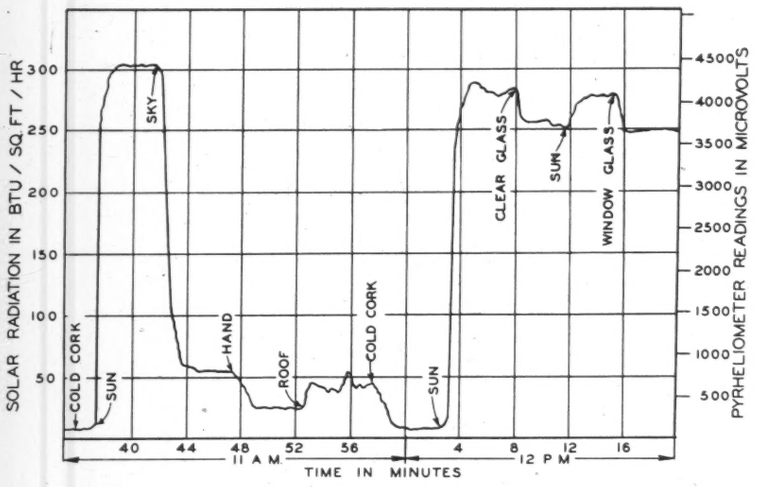


FIG 7—CURVE SHOWING PYRHELIOMETER INDICATION AND RADIATION INTENSITY FROM THE SUN, SKY, A PERSON'S HAND, A ROOF, AN ICE COLD SURFACE AND THE RADIATION TRANSMISSION THROUGH GLASS

localized heating or cooling, but not sufficiently to cause an appreciable increase in heat transfer from the under surface of the panels.

Nicholls heat flow meters,[6] *L* and *O*, were firmly fastened to the lower and the upper surfaces of the panels, and their edges were sealed to the panels with adhesive tape. These meters gave the instantaneous flows of heat for both the upper and lower surfaces. The upper meters were covered with black oilcloth, *K*, to protect them from the weather. To obtain uniform, natural surfaces, the upper oilcloth and the lower meter surface were painted a dead black with lampblack pigment thinned with weak shellac.

Thermocouples of No. 36 gage B & S copper-constantan wire were fixed in positions on the top surface of the weather proofing over each panel, between the top of the panels and the top meters, between the lower meters and the bottom surface of the panels, on the exposed lower surface of the lower meter, and 6 in., 18 in. and 30 in. below the lower surface of each panel. A thermocouple for observing air temperatures in the sun was located 6 in. above the center of the middle panel; another thermocouple for observing shaded air temperatures was put under a double-deck shield, *N*, Fig. 4; and extra thermocouples were located at all important points to serve as checks. An electrical cup anemometer, *M*, placed near the top of the panels recorded all air movement over the face of the panels.

[6] See Bibliography, *b*.

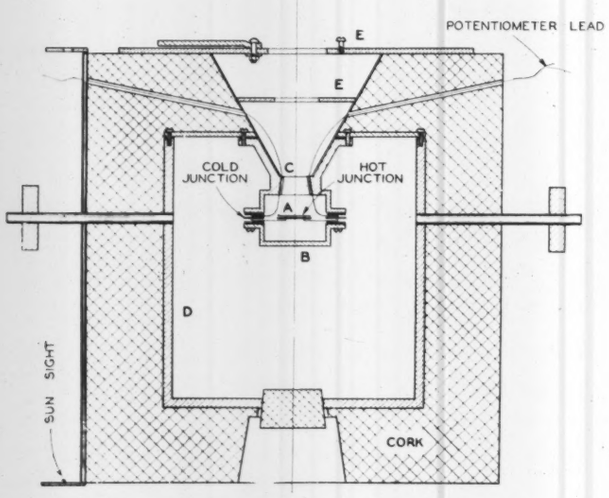


FIG. 6—CROSS SECTION OF THE LABORATORY PYRHELIOMETER

A pyrheliometer located near the set-up indicated the intensity of solar radiation throughout the day. This instrument, shown in the photograph, Fig. 5, and the drawing, Fig. 6, was designed and built at the A. S. H. V. E. Laboratory. It consisted of a smoked, dead-black, sensitive disk, *A*, containing thirty-one hot junctions of No. 40 B & S gage thermocouples, the cold junctions of which were located between the flanged joint of the cup, *B*. When the instrument was sighted on the sun, the sun shone onto the sensitive disk through the orifice, *C*, which was so designed that the sun entirely covered the thermocouple junctions in the disk, even when the instrument failed to point directly at the sun by an angle of two degrees. During a day, the cold junctions were held at practically a constant temperature because the insulated brass container, *D*, was filled with water, which has great heat capacity. Small changes in the temperature of the cold junctions did not affect readings of the instrument, since the temperature difference between the hot and cold junctions remained constant. The diaphragm shields, *E*, protected the sensitive disk from air currents and stray radiation. A trunnion mounting timed by a clock controlled the direction of the instrument so that when properly sighted in the morning, it followed the sun all day. Leads from the instrument went to either a recording or a precision potentiometer in the room below. Fig. 7 is a curve which gives some radiations read with the instrument. The scale at the right is in microvolts as recorded by the instrument. The scale at the left, in Btu per square foot per hour perpendicular to the sun's rays, corresponds to the record in microvolts obtained by calibrating the instrument with a Smithsonian silver disk pyrheliometer.

The instruments and apparatus for controlling air temperature within the chamber on the roof, for observing temperature, heat flows, wind velocity and in-

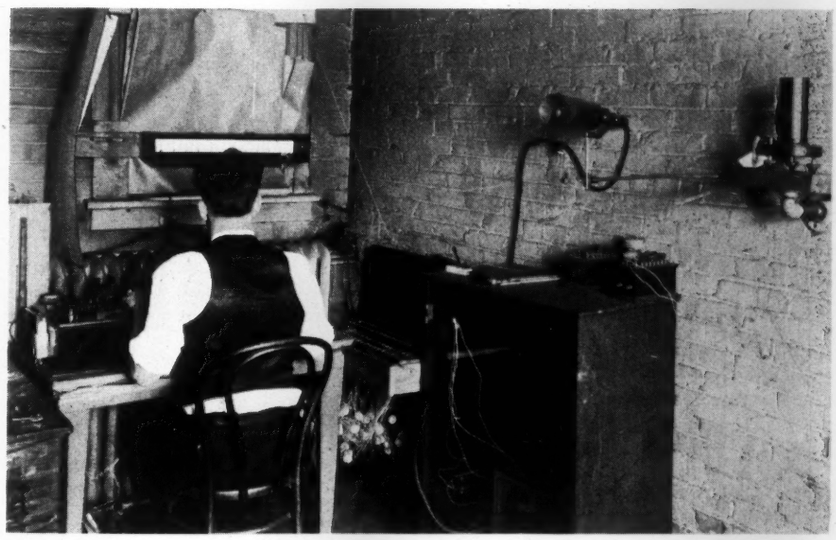
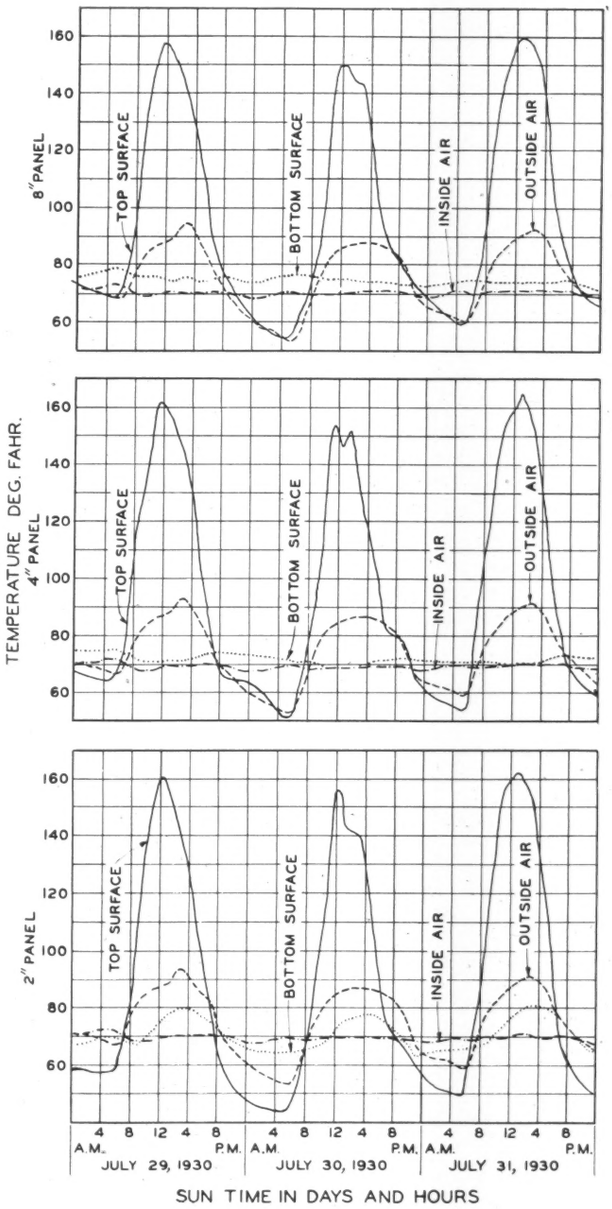


FIG. 8—APPARATUS AND INSTRUMENTS USED TO CONTROL TEMPERATURES AND TO OBSERVE TEMPERATURES, HEAT FLOWS, WIND VELOCITIES AND INTENSITY OF SOLAR RADIATION IN THE TEST SET-UP ON THE ROOF ABOVE

FIG. 9—CURVES SHOWING OBSERVED TEMPERATURES OF TOP AND BOTTOM SURFACES AND INSIDE AND OUTSIDE AIR FOR 2 IN., 4 IN., AND 8 IN. IRON AND CORK PANELS



tensity of solar radiation, were all located in a room directly below, and are shown in the photograph, Fig. 8.

This set-up allowed three panels to be tested simultaneously with heat flow meters and thermocouples so arranged that at any time readings could be had of the instantaneous heat flow into the top and out of the bottom surfaces, and of the temperature gradients through the panels subjected to varying weather conditions at the outer surface and constant air temperature below.

Test Operation and Results

Time observations were made in mean solar time, which for the latitude and longitude of Pittsburgh and for the days studied is 20 to 26 min behind Eastern Standard Time. All calculations involving time, and all references made to time, are in terms of mean solar, or sun time.

All tests were run continuously for 24 hours starting shortly before sunrise. If no test had been run the day before, the air temperature in the chamber was controlled at approximately 70 F for a preliminary time of twelve hours. This was done automatically by thermostatic control. During the 24 hours of the test period, the air temperature in the chamber was controlled manually at 69.6 F $\pm$ 0.2 deg. Half-hourly observations were made of the heat flow as given by the six heat meters, of the temperatures indicated by the thermocouples, of the wind velocity, the intensity of solar radiation, and the general weather conditions. Whenever weather conditions permitted tests were run continuously day after day. However, it was found that comparatively few days were sufficiently free from clouds or haze to permit the collection of satisfactory data.

The data used for analysis in this paper include the top surface temperature of the panel exposed to the air, the inner air temperature—always kept at 69.6 F—and the heat flow observed by the lower heat meter. To avoid errors introduced by edge losses, all temperatures were read at the center of the 18-in. square test section covered by the recording element of the heat flow meter. Fig. 9 shows the observed temperatures of the top and bottom surfaces of the 2-in., 4-in., and 8-in. laminated iron-cork panels and the inside and outside air tempera-

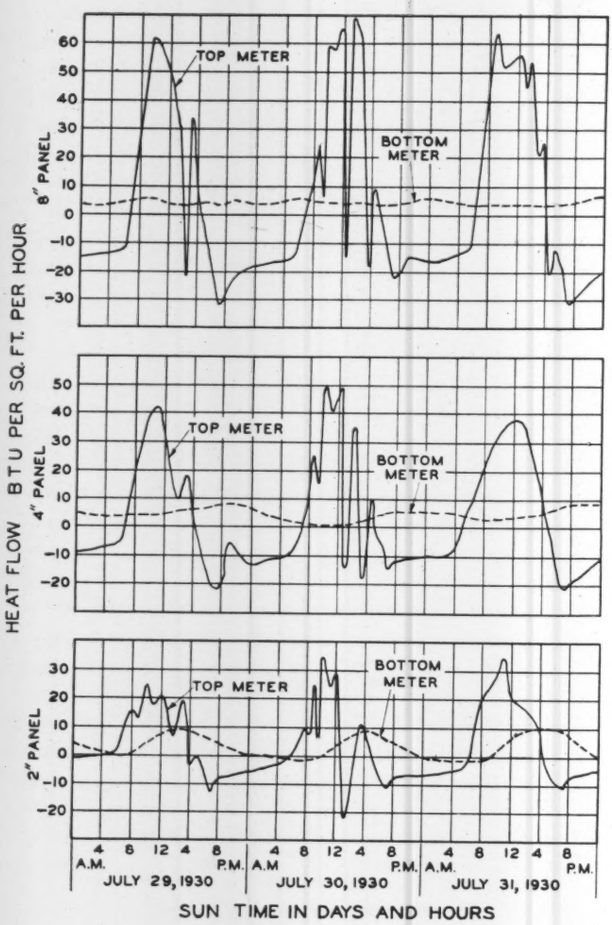
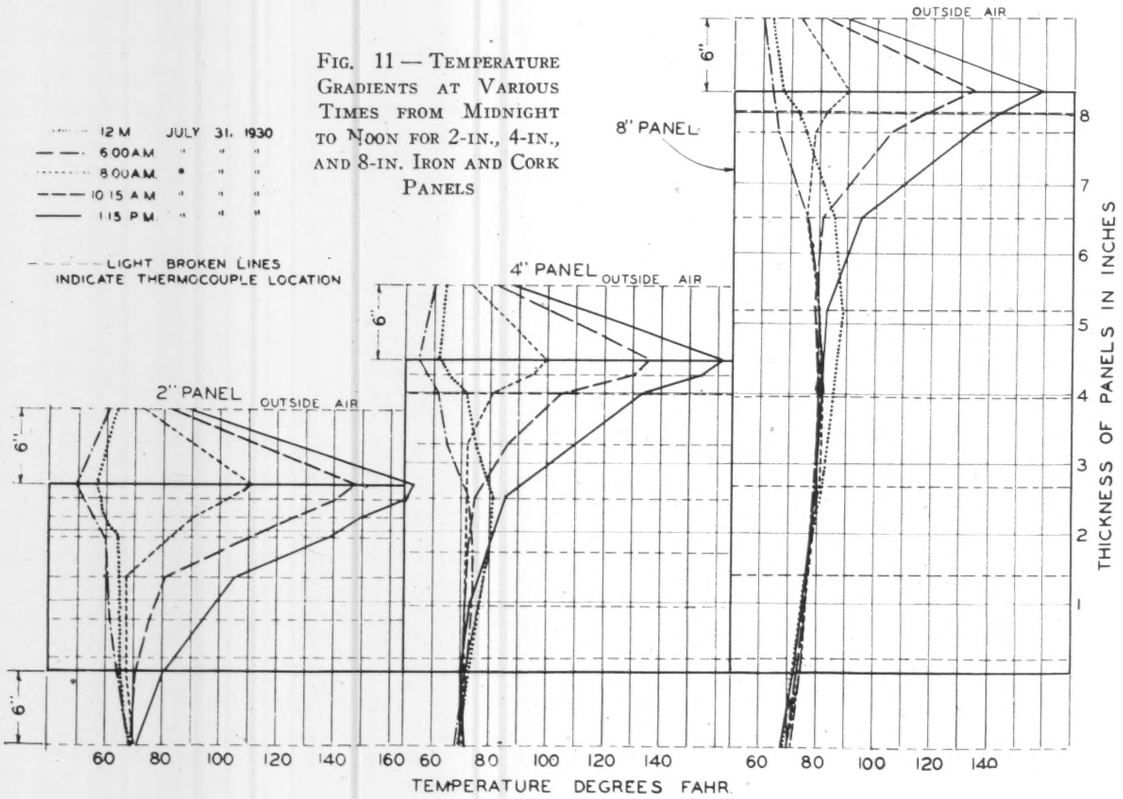


FIG. 10—CURVES GIVING OBSERVED HEAT FLOWS THROUGH TOP AND BOTTOM SURFACES FOR 2 IN., 4 IN. AND 8 IN. IRON AND CORK PANELS

tures; the four temperatures are plotted against time during three days, July 29, 30, and 31. Fig. 10 is a graph of the observed heat flows at the upper and lower surfaces for the same panels on the same days. Temperature gradients for one day through the iron and cork panels, as measured, are plotted in Fig. 11 for various times between midnight and noon, and are continued in Fig. 12 between noon and the following midnight. These give an interesting series of temperature gradient curves from the air above through the panel to the air below, which show just how the temperature wave progresses and how its amplitude is damped out as it penetrates to greater depths.

Fig. 13 gives the top surface temperature and the inner surface heat flow as observed on the pine panel for four days, July 25 to 28 inclusive. The result of a mathematical analysis of the heat flow through the lower surface is also given which will be discussed later. Similar data are given for three days on the concrete panel in Fig. 14. Figs. 15 and 16 give the solar radiation normal to the sun and normal to a horizontal surface for the days shown in Figs. 13 and 14. The intensity of solar radiation in Btu per square foot per hour on a surface perpendicular to the direction of the rays of the sun were obtained from the recorded chart of the A. S. H. V. E. Laboratory pyrheliometer, and the intensity of solar radiation on a horizontal surface was calculated from the recorded curve. Data similar to that in Figs. 13 and 14 are given for five days on the gypsum panel in Fig. 17, for three days on the cork-concrete (cork up) panel in Fig. 18, for two non-consecutive days on the concrete-cork (concrete up) panel in Fig. 19, and for one day each on the 2-in. and 4-in. iron-cork panels in Figs. 20 and 21. Figs. 22 and 23 give the temperatures of the top surface exposed to the

FIG. 11 — TEMPERATURE GRADIENTS AT VARIOUS TIMES FROM MIDNIGHT TO NOON FOR 2-IN., 4-IN., AND 8-IN. IRON AND CORK PANELS



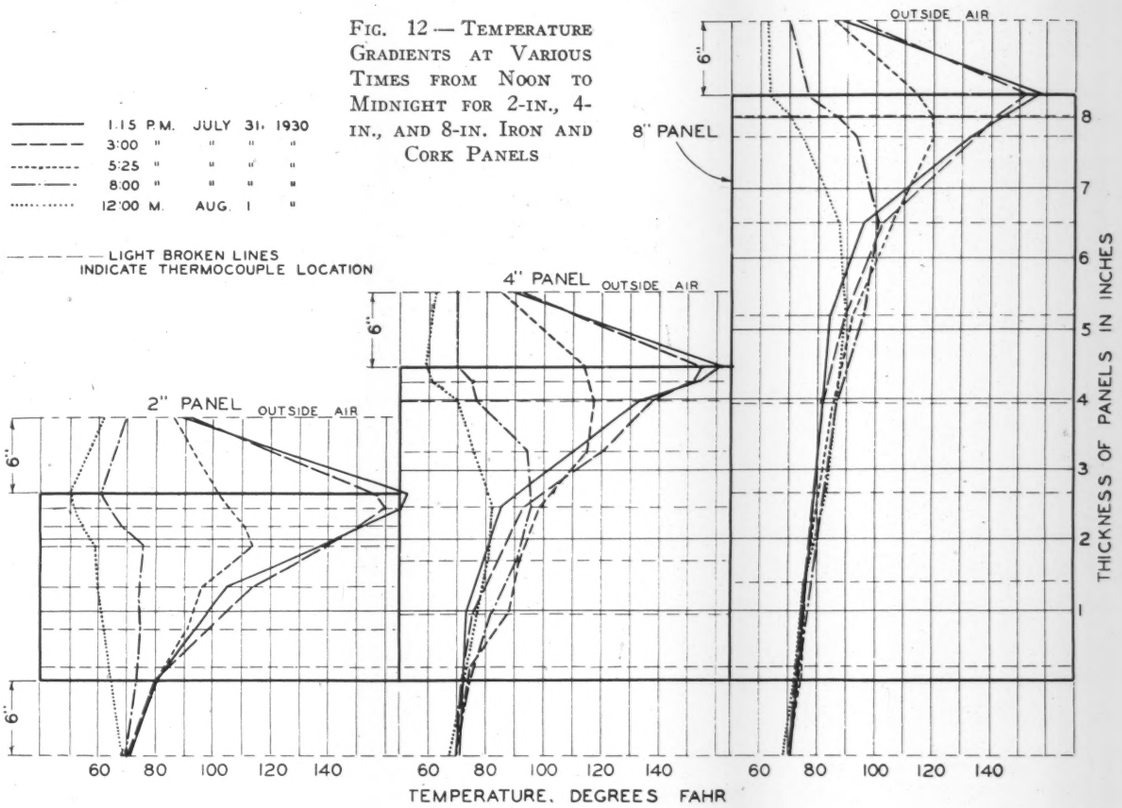


FIG. 12 — TEMPERATURE GRADIENTS AT VARIOUS TIMES FROM NOON TO MIDNIGHT FOR 2-IN., 4-IN., AND 8-IN. IRON AND CORK PANELS

atmosphere, the temperatures below the upper meter, the air temperatures above the panels, the temperatures of the lower surface exposed to the conditioned air, the heat flows through the lower surface, and other calculated curves which will be discussed later, for one day each on pine and concrete respectively. Figs. 24, 25 and 26 give the top surface temperature and the lower surface heat flows for an additional day each for pine, concrete and gypsum respectively.

The curves in Figs. 9 to 26 show that immediately after sunrise the top surface temperature begins to rise and that it continues rising until shortly after noon, when the temperature begins to recede rapidly until after sunset, from which point it continues to fall slowly until sunrise the next morning. The part of the curve from sunrise to sunset is a close approximation of a half cycle of a simple harmonic curve. The part of the curve from sunset to sunrise does not represent the other half cycle of the simple harmonic curve, but is shorter in time, because during the time of year represented by the data, nights are shorter than days. The amplitude of the night part of the cycle is not as great as for the day, and does not reach its trough midway between sunset and sunrise, but at a time very shortly before sunrise. This failure of the night half of the curve to reach a trough with amplitude equal to that at noontime may be accounted for by the lack of negative solar radiation at night. The heat flow through the lower surface gives a curve similar in shape to the top surface temperature, but lagging behind it by an amount

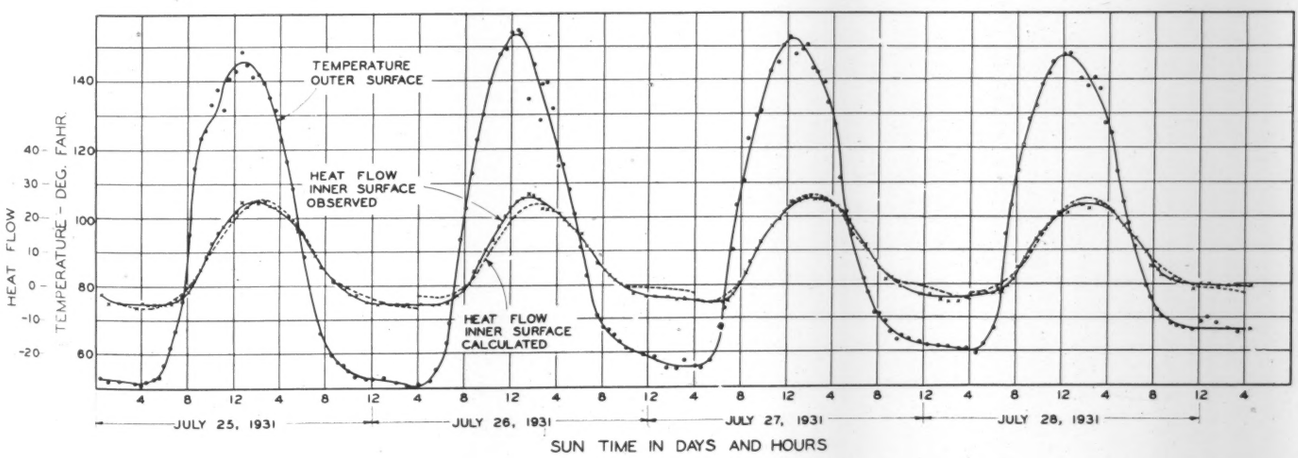


FIG. 13—CURVES FOR 2-IN. PINE PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND AS CALCULATED

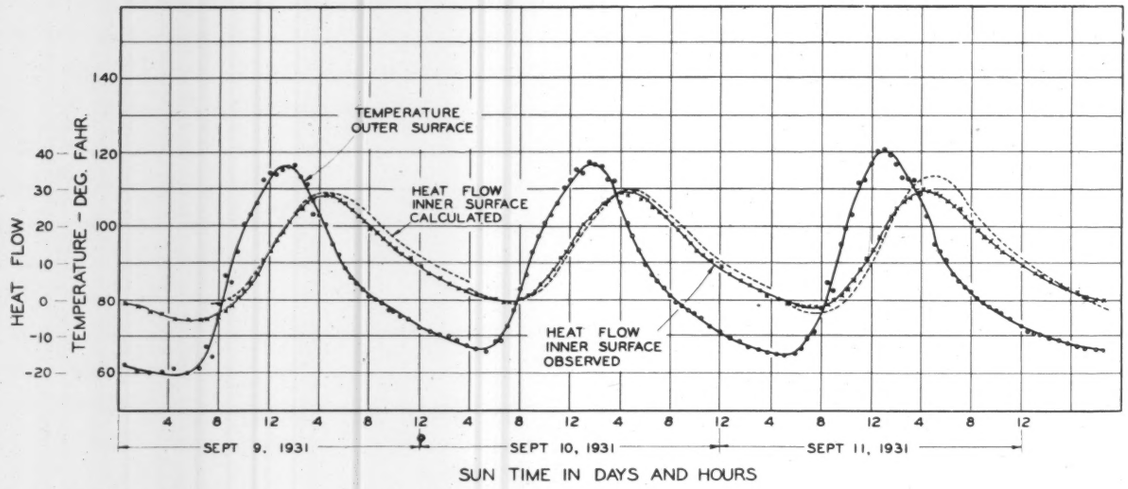


FIG. 14—CURVES FOR 6-IN. CONCRETE PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND AS CALCULATED

TABLE 3—CHARACTERISTICS OF TOP SURFACE TEMPERATURES AND LOWER SURFACE HEAT FLOWS

PANEL	DATE 1930	OUTSIDE AIR TEMPERATURE MAX. °F.	MAX. TIME	MIN. °F.	MIN. TIME	OUTER SURFACE TEMPERATURE BEGAN TO RISE A.M.	CREST TIME	TROUGH TIME	CREST TEMP.	TROUGH TEMP.	TEMP. RANGE	INNER SURFACE HEAT FLOW CREST TIME	TROUGH TIME	CREST BTU	TROUGH BTU	BTU RANGE	LAG BETWEEN CREST OF TEMP AND CREST OF HEAT FLOW IN HOURS
2" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	5:40	12:15 P.M.	5:10 A.M.	160.5	57.5	103.0	2:30 P.M.	6:30 A.M.	9.0	0.0	9.0	2 Hrs. 15 Min
" " "	" 30	87.0	2:15 P.M.	53.6	5:10 "	5:00	12:15 "	4:50 "	156.0	43.5	112.5	4:00 "	7:30 "	9.0	2.5	6.5	3 " 0 "
" " "	" 31	85.5	2:27 "	59.4	5:13 "	5:10	12:50 "	5:10 "	162.0	49.5	112.5	4:30 "	6:30 "	10.5	3.0	7.5	3 " 40 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:05 "	5:00	12:00 "	5:00 "	159.3	42.5	116.8	3:05 "	5:10 "	11.0	-3.5	14.5	3 " 05 "
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:25	12:30 "	5:25 "	153.0	40.2	112.8	2:30 "	7:00 "	11.2	-3.5	14.7	2 " 0 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	6:05	1:25 "	5:25 "	133.5	38.8	94.7	2:20 "	3:40 "	10.3	-4.5	14.8	0 " 55 "
2" Iron - Cork	" 19	88.0	11:34 A.M.	54.0	7:09 "	7:15	1:00 "	5:50 "	135.3	44.2	91.1	3:00 "	1:00 "	6.9	-3.4	10.3	2 " 0 "
4" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	4:30	11:30 A.M.	4:30 A.M.	162.5	64.5	98.0	9:00 P.M.	4:30 A.M.	8.0	3.5	4.5	9 " 30 "
" " "	" 30	87.0	2:15 "	53.6	5:10 "	5:30	11:30 "	5:00 "	153.0	42.0	111.0	8:00 "	11:30 "	5.5	0.5	5.0	8 " 30 "
" " "	" 31	85.5	2:27 "	59.4	5:13 "	5:00	1:15 P.M.	5:00 "	164.5	44.0	120.5	10:00 "	9:00 "	8.5	3.0	5.5	8 " 45 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:05 "	5:00	12:40 "	5:00 "	149.1	46.0	103.1	5:40 "	8:40 "	6.5	1.7	4.8	5 " 0 "
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:50	12:50 "	5:30 "	147.7	44.1	103.6	6:40 "	12:50 P.M.	5.2	-1.0	6.2	5 " 40 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	6:00	12:40 "	6:00 "	135.0	44.8	90.2	8:20 "	5:30 "	4.8	3.3	1.5	
4" Iron - Cork	" 19	88.0	11:34 A.M.	54.0	7:09 "	6:35	12:00 "	5:50 "	129.0	47.4	81.6	6:40 "	2:50 "	4.7	1.1	5.8	6 " 40 "
8" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	4:50	11:20 A.M.	4:50 A.M.	157.0	68.5	88.5	9:30 A.M.	7:00 P.M.	5.5	3.5	2.0	
" " "	" 30	87.0	2:15 "	53.6	5:10 "	4:15	12:00 P.M.	4:15 "	149.5	44.5	105.0	6:30 "	6:00 "	6.0	3.5	2.6	19 " 10 "
" " "	" 31	85.6	2:27 "	59.4	5:13 "	5:30	1:00 "	5:00 "	159.5	49.0	110.5	12:00 P.M.	4:00 "	7.0	3.5	3.5	24 " 0 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:05 "	5:30	1:05 "	5:30 "	146.5	48.6	97.9	12:22 "	5:00 "	4.8	3.3	1.5	
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:35	12:19 "	5:45 "	144.0	47.6	96.4	8:40 A.M.	1:34 "	4.5	2.0	2.5	15 " 35 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	5:57	1:00 "	5:57 "	132.8	50.2	82.6	10:30 "	8:20 "	7.7	3.2	4.5	
8" Iron - Cork	" 19	88.0	1:22 "	54.0	5:35 "	7:18	1:25 "	7:00 "	127.0	50.1	66.9	6:20 "	4:51 "	6.5	-0.9	7.4	17 " 20 "
2" Pine	1931 July 25	86.6	2:30 P.M.	56.2	3:25 A.M.	5:00	12:50 P.M.	3:40 A.M.	146.0	51.5	94.5	2:00 P.M.	6:00 A.M.	25.0	-5.0	30.0	1 " 10 "
" "	" 26	94.4	1:20 "	56.1	4:00 "	4:50	12:40 "	3:40 "	153.5	50.7	102.8	1:40 "	5:20 "	26.0	-4.8	30.8	1 " 0 "
" "	" 27	97.5	1:45 "	60.8	4:30 "	5:10	12:30 "	3:00 "	152.5	56.0	96.5	2:10 "	5:20 "	25.5	-5.2	30.7	1 " 40 "
2" Pine	" 28	96.6	3:15 "	65.5	3:45 "	4:30	12:15 "	4:10 "	147.0	60.2	86.8	2:15 "	2:00 "	24.0	-3.8	27.8	2 " 0 "
6" Concrete	July 25	86.6	2:30 P.M.	56.2	3:25 A.M.	5:00	1:30 P.M.	4:30 A.M.	125.0	60.0	65.0	5:00 P.M.	7:00 A.M.	38.2	-5.0	43.2	3 " 30 "
" "	" 26	94.4	1:20 "	56.1	4:00 "	5:00	12:45 "	4:45 "	125.8	60.3	65.5	4:35 "	6:50 "	37.5	-4.0	41.5	3 " 50 "
" "	" 27	97.5	1:45 "	60.8	4:30 "	5:00	1:50 "	5:00 "	128.5	65.0	63.5	4:50 "	6:30 "	40.8	0.2	40.6	3 " 0 "
" "	" 28	96.6	3:15 "	65.5	3:45 "	5:30	1:40 "	5:00 "	125.3	68.3	57.0	4:15 "	6:30 "	41.0	1.2	39.8	2 " 35 "
" "	Sept. 7	82.2	1:30 "	50.5	4:35 "	5:50	1:20 "	5:25 "	118.2	55.2	63.0	4:05 "	7:15 "	30.5	-6.5	37.0	2 " 45 "
" "	" 8	82.8	3:30 "	51.7	6:00 "	6:00	3:00 "	5:15 "	96.0	55.0	41.0	5:00 "	7:20 "	15.0	-9.0	24.0	2 " 0 "
" "	" 9	92.9	1:30 "	56.4	1:30 "	5:30	1:15 "	5:00 "	116.0	59.5	56.5	4:30 "	5:40 "	28.0	-6.0	34.0	3 " 15 "
" "	" 10	97.9	3:00 "	65.5	5:15 "	5:40	1:30 "	4:50 "	117.0	66.5	50.5	4:10 "	7:20 "	29.0	-5.0	29.5	2 " 40 "
6" Concrete	" 11	95.9	3:00 "	64.7	5:15 "	5:45	1:10 "	5:00 "	120.0	65.0	55.0	4:00 "	7:10 "	29.5	-2.0	31.5	2 " 50 "
4" Gypsum	Sept. 7	82.2	1:30 P.M.	50.5	4:35 A.M.	5:30	1:00 P.M.	5:30 A.M.	138.5	46.0	92.5	3:15 P.M.	7:00 A.M.	18.0	-5.8	23.8	2 " 15 "
" "	" 8	82.8	3:30 "	51.7	6:00 "	5:45	3:20 "	5:30 "	116.0	48.0	68.0	5:00 "	6:45 "	9.5	-6.2	15.7	1 " 40 "
" "	" 9	92.9	1:30 "	56.4	1:30 "	6:00	1:00 "	1:50 "	132.4	53.5	78.9	4:10 "	5:00 "	16.5	-3.2	19.7	3 " 10 "
" "	" 10	97.9	3:00 "	65.5	5:15 "	5:50	1:35 "	5:30 "	132.0	61.5	70.5	4:00 "	7:00 "	18.5	-1.4	19.9	2 " 25 "
4" Gypsum	" 11	95.9	3:00 "	64.7	5:15 "	6:30	1:00 "	5:00 "	140.6	60.5	80.1	4:20 "	7:00 "	8.3	-3.0	11.3	3 " 20 "
4" Cork - Concrete	Sept. 7	82.2	1:30 P.M.	50.5	4:35 A.M.	6:00	12:40 P.M.	5:30 A.M.	139.0	43.0	96.0	4:00 P.M.	7:30 A.M.	12.0	-1.8	13.8	3 " 20 "
" " "	" 8	82.8	3:30 "	51.7	6:00 "	5:40	2:50 "	5:20 "	121.0	45.0	76.0	5:10 "	6:50 "	6.4	-3.9	10.3	2 " 20 "
" " "	" 9	92.9	1:30 "	56.4	1:30 "	2:00	12:35 "	2:00 "	135.8	51.8	84.0	4:10 "	5:30 "	11.8	-2.8	14.8	3 " 35 "
" " "	" 10	97.9	3:00 "	65.5	5:15 "	5:00	1:00 "	4:30 "	136.5	60.0	76.5	4:00 "	7:00 "	12.0	-1.0	13.0	3 " 0 "
4" Cork - Concrete	" 11	95.9	3:00 "	64.7	5:15 "	6:00	12:50 "	4:30 "	141.0	59.2	81.8	4:10 "	6:00 "	12.0	-1.0	13.0	3 " 20 "
4" Concrete - Cork	July 25	86.8	2:30 P.M.	56.2	3:25 A.M.	5:00	2:00 P.M.	4:50 A.M.	137.5	55.2	82.3	4:05 P.M.	6:30 A.M.	15.9	-2.7	18.6	2 " 05 "
" " "	" 26	94.4	1:20 "	56.1	4:00 "	5:30	1:50 "	5:00 "	142.0	57.0	85.0	4:00 "	5:40 "	15.8	-3.0	18.8	2 " 10 "
" " "	" 27	97.5	1:45 "	60.8	4:30 "	5:20	2:00 "	5:20 "	141.0	63.2	77.8	4:00 "	6:00 "	17.5	-1.2	18.7	2 " 0 "
4" Concrete - Cork	" 28	96.6	3:15 "	65.5	3:45 "	5:30	1:50 "	5:10 "	137.0	67.0	70.0	3:00 "	6:40 "	16.9	-0.2	17.1	1 " 10 "

depending upon the panel studied. Table 3 gives timing and characteristics of the top surface temperatures and the lower surface heat flows for different days and panels.

Comparisons between the calculated and observed heat flows for the eight panels are shown for different periods of the day in Table 4. The calculated heat flows were based on the conductivities and inner film conductances given in Table 1, an outer film conductance of 2.0, and the average difference in temperature between the inside and outside air. Direct comparison between panels can be made only for those studied on the same day. It will be observed that for any panel the heat flow reaches a maximum sometime after noon and then recedes. The calculated heat flow shows a similar cycle, though not necessarily timed the same, and with widely different magnitudes. All panels give maximum heat penetration into the interior during the period, 3 p.m. to 4 p.m.,

TABLE 4—OBSERVED AND CALCULATED HEAT FLOW

Time	No. of Hours	HEAT FLOW															
		2" Pine July 27, 1931		6" Concrete Sept. 10, 1931		Gypsum Sept. 10, 1931		Cork - Concrete Sept. 10, 1931		Concrete - Cork July 27, 1931		2" Iron - Cork July 31, 1930		4" Iron - Cork July 31, 1930		8" Iron - Cork July 31, 1930	
		Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.
6-7 A.M.	1	-5.5	0.6	0	-1.4	-1.0	-0.6	0	-0.5	-1.5	0.5	-1.7	-0.7	2.4	-0.8	3.0	-0.7
9-10 "	1	12.0	4.4	2.5	11.1	1.0	5.1	1.0	4.4	2.5	3.6	2.5	1.6	1.5	1.6	3.4	1.5
12-1 P.M.	1	23.0	6.4	18.0	13.9	11.0	6.3	8.5	5.6	11.5	5.2	8.5	2.5	2.0	2.6	3.5	2.4
3-4 "	1	24.5	6.5	28.5	15.0	17.0	6.8	11.5	6.0	17.5	5.3	10.5	2.6	4.7	2.7	3.7	2.5
6-7 "	1	12.5	3.3	25.0	8.4	13.5	3.8	10.0	3.4	14.0	2.7	8.0	0.7	6.5	0.8	3.8	0.7
9-10 "	1	1.5	0.9	15.0	1.6	4.0	0.7	5.0	0.6	7.5	1.8	2.0	-0.1	7.9	-0.1	4.0	-0.1
12-1 A.M.	1	-3.0	-0.4	7.5	-1.2	1.0	-0.6	2.0	-0.5	2.0	-0.4	-1.0	-0.6	6.0	-0.7	4.5	-0.6
3-4 "	1	-4.0	-1.1	1.5	-2.4	-1.0	-1.1	.0	-1.0	0.5	-0.9	-1.4	-1.2	4.0	-1.2	4.0	-1.1
9 A.M.-5 A.M.	8	153.5	52.0	146.3	104.0	81.6	47.2	60.2	41.6	88.7	42.4	65.1	19.0	23.3	19.4	24.1	17.8
8 P.M.-11 P.M.	3	5.6	2.9	48.0	4.6	19.2	2.1	18.7	1.8	15.7	1.6	4.1	0.4	14.1	-2.7	10.5	-0.3

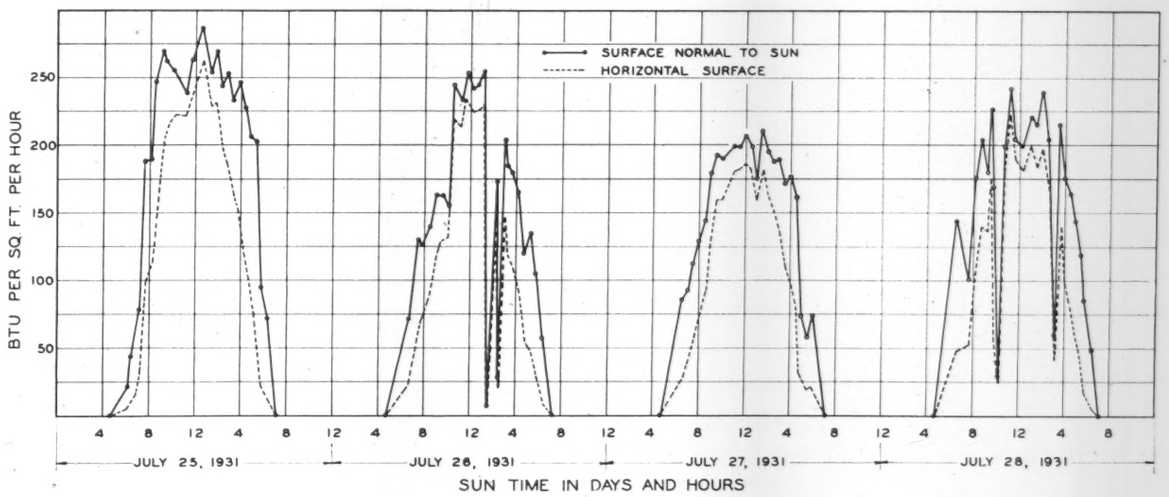

SURFACE NORMAL TO SUN
HORIZONTAL SURFACE
BTU PER SQ. FT. PER HOUR
250
200
150
100
50
4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8
JULY 25, 1931
JULY 26, 1931
JULY 27, 1931
JULY 28, 1931
SUN TIME IN DAYS AND HOURS

FIG. 15—CURVES GIVING SOLAR INTENSITY NORMAL TO SUN AND ON HORIZONTAL SURFACE

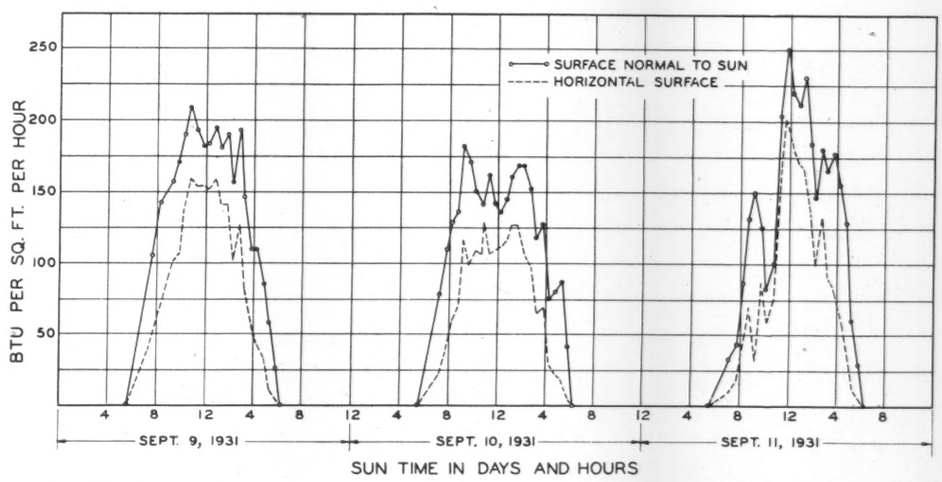

SURFACE NORMAL TO SUN
HORIZONTAL SURFACE
BTU PER SQ. FT. PER HOUR
250
200
150
100
50
4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8 12 4 8
SEPT. 9, 1931
SEPT. 10, 1931
SEPT. 11, 1931
SUN TIME IN DAYS AND HOURS

FIG. 16—CURVES GIVING SOLAR INTENSITY NORMAL TO SUN AND ON HORIZONTAL SURFACE

excepting those of iron-cork construction, which give maximum rates from 3 p.m. to 9 p.m. for the 2-in. and 4-in., and after midnight for the 8-in. panel. It will be observed that the calculated heat flow for the day and evening periods, 9 a.m. to 5 p.m., and 8 p.m. to 11 p.m., is always less than the observed heat flow for the same periods. At some times this discrepancy amounts to an observed value of approximately three

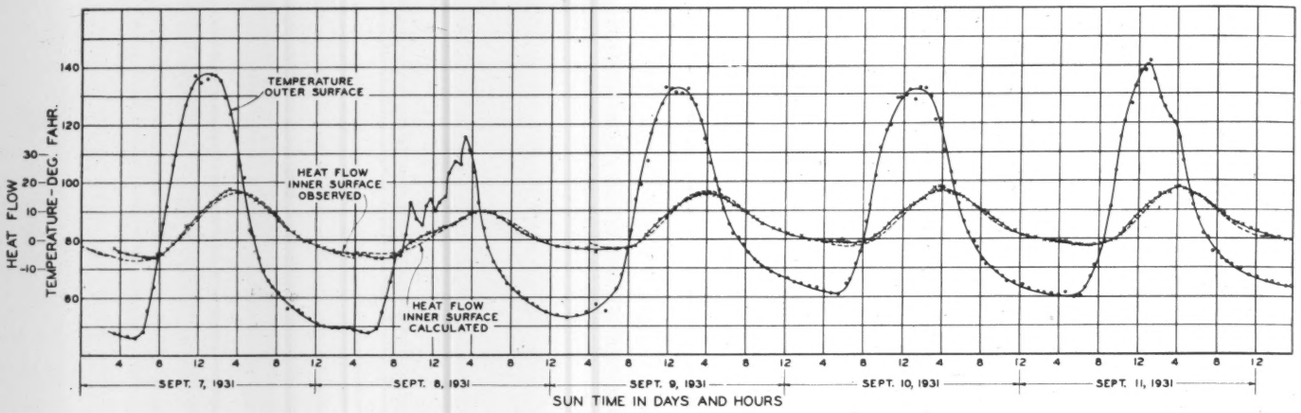
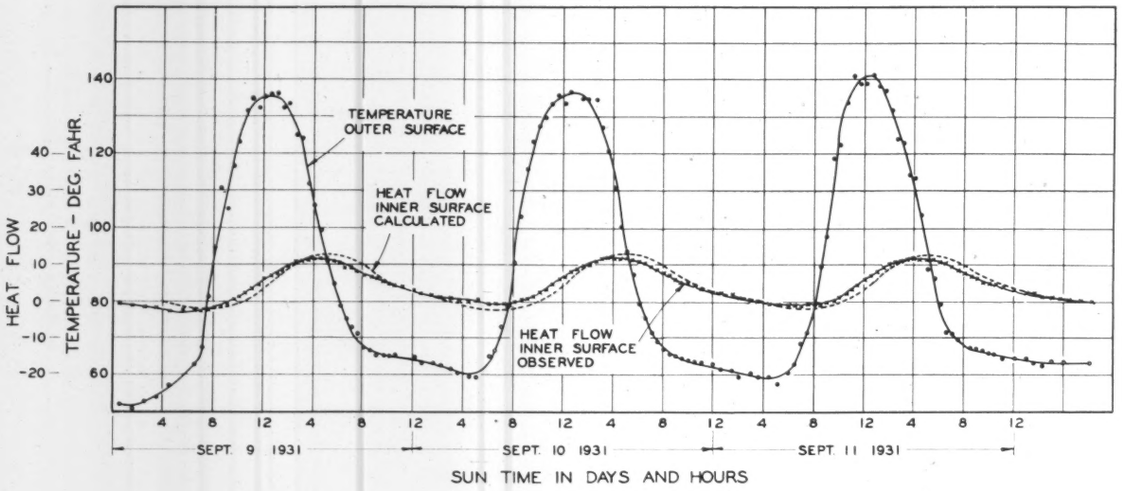


FIG. 17—CURVES FOR 4-IN. GYPSUM PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED



THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

FIG. 18—CURVES FOR 4-IN. CORK-CONCRETE PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW

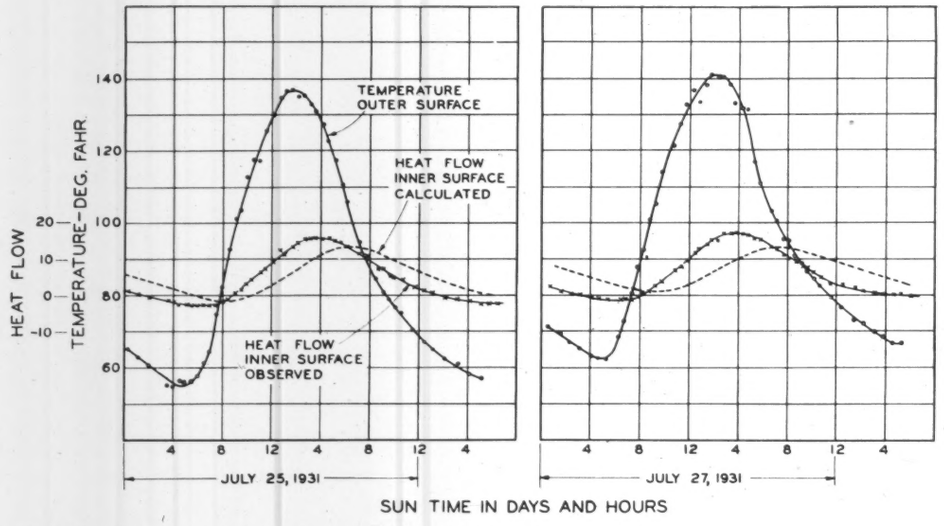


FIG. 19—CURVES FOR 4-IN. CONCRETE-CORK PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

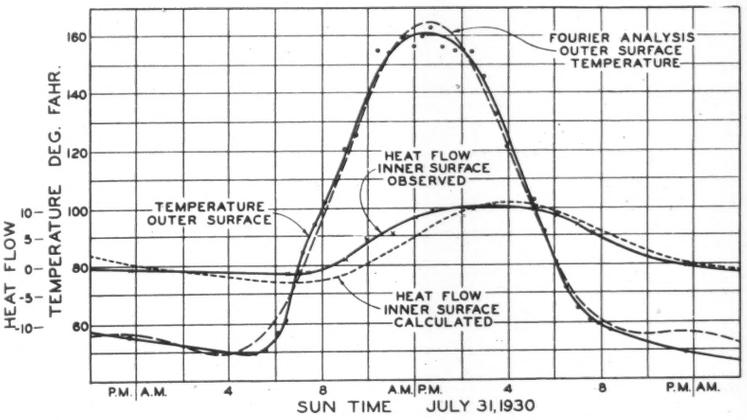

FOURIER ANALYSIS OUTER SURFACE TEMPERATURE
HEAT FLOW INNER SURFACE OBSERVED
TEMPERATURE OUTER SURFACE
HEAT FLOW INNER SURFACE CALCULATED
HEAT FLOW
TEMPERATURE DEG. FAHR.
P.M. A.M. 4 8 A.M. P.M. 4 8 P.M. A.M.
SUN TIME JULY 31, 1930

FIG. 20.—CURVES FOR 2-IN. IRON-CORK PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

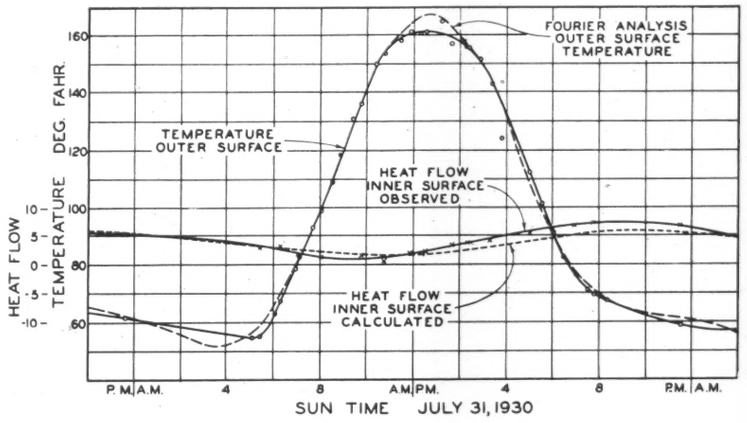

FOURIER ANALYSIS OUTER SURFACE TEMPERATURE
TEMPERATURE OUTER SURFACE
HEAT FLOW INNER SURFACE OBSERVED
HEAT FLOW INNER SURFACE CALCULATED
HEAT FLOW
TEMPERATURE DEG. FAHR.
P.M. A.M. 4 8 A.M. P.M. 4 8 P.M. A.M.
SUN TIME JULY 31, 1930

FIG. 21—CURVES FOR 4-IN. IRON-CORK PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

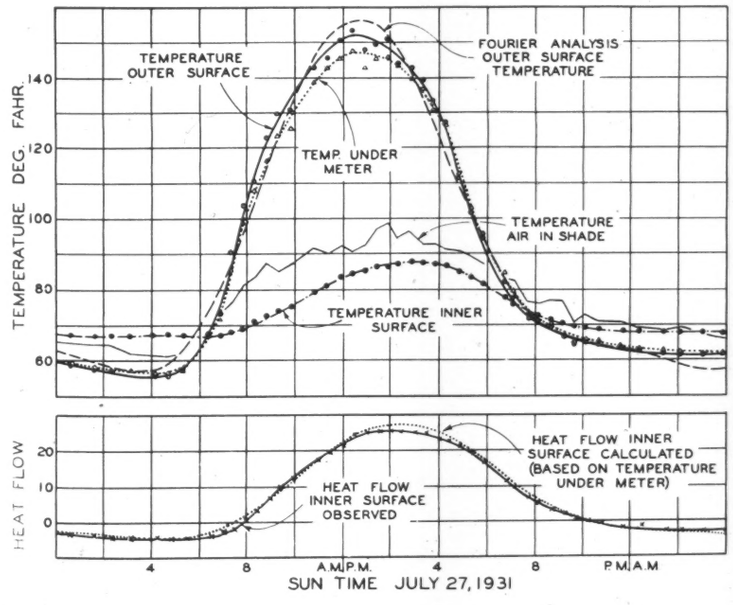

TEMPERATURE OUTER SURFACE
FOURIER ANALYSIS OUTER SURFACE TEMPERATURE
TEMP. UNDER METER
TEMPERATURE AIR IN SHADE
TEMPERATURE INNER SURFACE
TEMPERATURE DEG. FAHR.
HEAT FLOW INNER SURFACE CALCULATED (BASED ON TEMPERATURE UNDER METER)
HEAT FLOW INNER SURFACE OBSERVED
HEAT FLOW
4 8 A.M. P.M. 4 8 P.M. A.M.
SUN TIME JULY 27, 1931

FIG. 22—CURVES FOR 2-IN. PINE PANEL SHOWING TEMPERATURES AND HEAT FLOWS AS OBSERVED AND CALCULATED

times the calculated value for the 2-in. pine and the 2-in. iron and cork panels during the day period, and to an observed value ten times greater than the calculated value for the 6-in. concrete panel during the evening period.

Since the value of the conductivity is important in the analysis of the flow of heat, its accurate determination is necessary. For the earlier part of the work, the conductivities used were taken from accepted data in the A. S. H. V. E. GUIDE, but it was soon apparent that a check on these evaluated values would be desirable. By holding the air in the chamber at a constant temperature and covering the upper surface with a deep bed of 32 F slush ice and water, an excellent determination of the conductivities of the panels was made with them in place. The calibrated Nicholls heat flow meters on the upper and lower surfaces gave the heat flow into and out of the lower and upper surfaces of the panels, in addition to a check of possible heat losses through the edges. A frame, 6 in. deep, was built up on the outer edges of the panels to hold the ice in position. The respective heat flows and the temperatures of the different surfaces of the panels were recorded when the rate of flow became constant. The tests for the determination of conductances and conductivities were continued for a length of time sufficient to assure a balanced steady state condition of heat flow. From the temperatures, rates of heat flow, and thickness of the panel, it was a simple matter to work out the conductivities by the ordinary straight line flow formulae. These conductivities were of particular advantage, as they were run only a few days before the cyclic tests were made; consequently there was little time for the conductivities to change by ageing or drying.

The specific heats used for the first wave flow calculations were originally handbook data, but these were later checked by a rather unique method developed at the Laboratory. A pulverized sample of the material held in a sealed glass test tube was cooled to a constant temperature of 32 F by completely covering it with slush ice over night. Water at a temperature slightly above that of the room was kept agitated in the cup of a student calorimeter while the cooled sample of material was quickly added to the cup. All masses and temperatures were chosen so that the resulting mixture had a temperature the same as that of the room. Temperature observations were made of the mixture of sample and water with an accurate thermometer, and curves for cooling corrections were drawn. The specific heat of the unknown sample was determined from the weight, temperatures and constants for the calorimeter by the usual calculations. The fact that the sample was cooled instead of heated before being mixed with the water in the calorimeter greatly added to the accuracy of

the method, since the ice bath gave a very constant temperature, insuring an accurate temperature determination of the sample which would be otherwise hard to get. By mixing 32-F samples of the materials with 32-F water, an indication could be obtained if the heat of solution was affecting the results. The heat of solution did not enter into the calculations for the materials studied.

The lower surface film transmittance coefficient used in the mathematical analysis of heat flow was determined from the measured heat flow and the temperatures of the inner surface and the air 6 in. below the surface. Since the lower surfaces of all panels exposed to the air were painted with lampblack and shellac, the same coefficient applies to all. Analysis of many results for the lower surface gave 1.9 Btu per hour per square foot per degree temperature difference, which value is used in this paper. Many other data resulting from the study are available for analysis of the film transmittance for the lower surface and for the upper surface with still and moving air as measured at night, on clear and cloudy days, and for other conditions.

Mathematical Analysis of Periodic Heat Flow Through Roof

It is desired to develop as simple a method as possible for use in determining the rate of heat flow into an enclosure, using outside weather conditions and the physical properties of the structure. Such method is available for steady state conditions where the outside and inside air or surface temperatures have remained constant for a sufficient length of time for the rate of heat flow to become constant. It appears, however, that there is no method available for the case considered in this paper, where the outside source of heat and the outside surface temperature vary as a harmonic function of time.

It has already been pointed out that the problem of starting with the outside source of heat from the sun and the outside air temperature is infinitely more complicated than the problem of starting from the outside surface temperature; this paper is limited to a solution of the latter part of the complicated problem, leaving for future analysis the problem of arriving at the outside surface temperature using as factors the intensity of solar radiation, the outside weather conditions, and the physical properties of the structure. Stated more specifically, the problem is to take the outside surface temperature curve for a twenty-four hour period, as plotted in any of the Figs. 9 to 26, the thickness, conductivity, density and specific heat of the panel, the inside air temperature, and the inside surface film resistance; and from these to determine by calculation the rate at any intant of heat flow from the inside surface to the air. The nearest approach to a solution

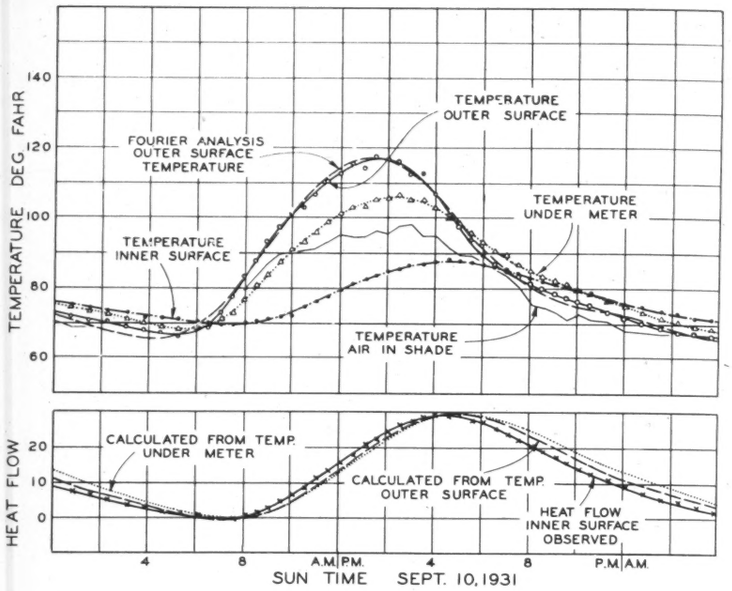


Fig. 23—Curves for 6-in. Concrete Panel Showing Temperatures and Heat Flows as Observed and Calculated

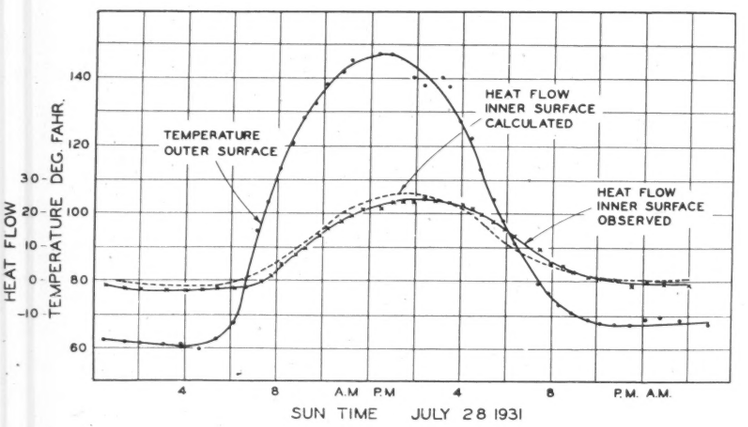


Fig. 24—Curves for 2-in. Pine Panel Showing Outer Surface Temperature, Observed Inner Surface Heat Flow, and Inner Surface Heat Flow as Calculated by the Empirical Solution

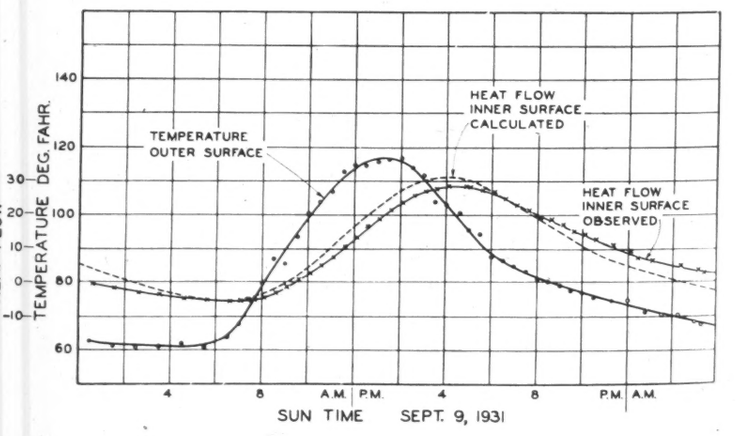


Fig. 25—Curves for 6-in. Concrete Panel Showing Outer Surface Temperature, Observed Inner Surface Heat Flow, and Inner Surface Heat Flow as Calculated by the Empirical Solution

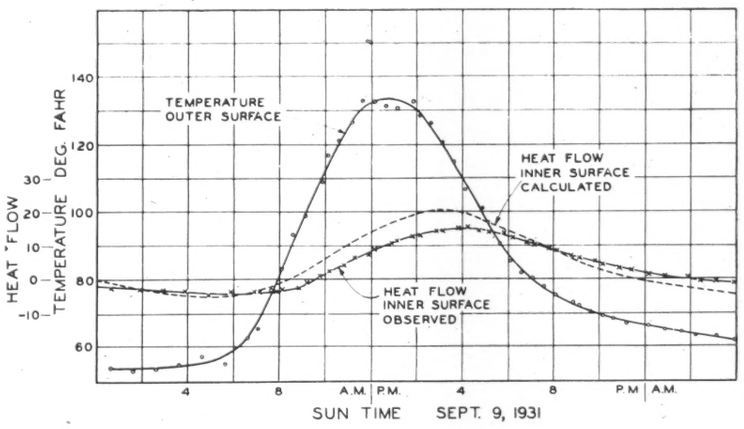


FIG. 26—CURVES FOR 4-IN. GYPSUM PANEL SHOWING OUTER SURFACE TEMPERATURE, OBSERVED INNER SURFACE HEAT FLOW, AND INNER SURFACE HEAT FLOW AS CALCULATED BY THE EMPIRICAL SOLUTION

of this has been in the work of Ingersoll and Zobel,[7] concerning the rate of heat transfer at any point within an infinitely thick solid, resulting from the application of simple harmonic temperature variation at the outside surface. While the Ingersoll and Zobel solution serves the problem in hand better than the assumption of steady state conditions, it is not adaptable to the needs of the air-conditioning engineer.

Mathematically, the solution of the following equation of heat flow is desired:

$$\frac{d\theta}{dt} = h^2 \frac{d^2\theta}{dx^2} \qquad (2)$$

which will satisfy the boundary conditions:

$\theta = \theta_o \sin \omega t$ at $x = 0$, and

$$k \frac{d\theta}{dx} + \epsilon\theta = 0 \quad \text{at } x = l \qquad (4)$$

For the sake of simplicity, it is assumed that the average temperature over a complete cycle is zero on both sides of the wall. No generality is lost by this assumption, for when the average temperatures on either side of the wall differ from zero, the steady flow of heat due to these average temperatures may be calculated and the two solutions added.

For solving the oscillatory case, the terms are defined as follows:

k = coefficient of thermal conductivity of slab
ρ = density of slab
c = specific heat of slab
ϵ = ratio of the heat flow per unit area from the slab surface to the air, to the difference in temperature between this surface and air. Assumed constant.
l = thickness of the slab
$\omega = \frac{2\pi}{T}$ where T is the period of the temperature oscillations

$$h = \sqrt{\frac{k}{c\rho}}$$

$$\beta = \frac{1}{h}\sqrt{\frac{\omega}{2}} = \sqrt{\frac{c\rho\omega}{2k}}$$

$$f = \frac{\epsilon}{k\beta} = \frac{\epsilon\sqrt{2}}{\sqrt{c\rho\omega k}}$$

The solution is

$$\theta = c_1 e^{-\beta x} \sin(\omega t + \gamma_1 - \beta x) + c_2 e^{\beta(x-2l)} \sin[\omega t + \gamma_2 + \beta(x-2l)] \qquad (5)$$

See Bibliography, c

The first term in Equation 5 represents the direct wave and the second term, a wave reflected from the inner surface. In the general case, there may be multiple reflections as in the optical case, and so each term must represent the sum of an infinite series of waves. However, where dealing with insulating materials, the amplitude of the temperature wave is so rapidly damped out that only the first reflection from the inner surface is of importance. Equation 5 may be written:

$$\theta = \frac{\theta_o}{B} e^{-\beta x} \sin(\omega t + \alpha - \beta x) + \frac{A\theta_o}{B} e^{\beta(x-2l)} \sin[\omega t + \alpha + \sigma + \beta(x-2l)] \qquad (6)$$

where

$$A = +\sqrt{\frac{(1-f)^2+1}{(1+f)^2+1}}$$

$$\sigma = \cos^{-1}\frac{2-f^2}{\sqrt{4+f^4}} \qquad 0 \leqq \sigma \leqq \pi$$

$$\alpha = \sin^{-1}\frac{A}{B} e^{-2\beta l} \sin(\sigma - 2\beta l) \qquad (-\pi < \alpha < \pi)$$

$$B = \sqrt{1 + A^2 e^{-4\beta l} + 2Ae^{-2\beta l}\cos(\sigma - 2\beta l)} \qquad (7)$$

In practice, $e^{-2\beta l}$ will be small, and $e^{-4\beta l}$ will be very small, while A must lie between 0.546 and + 1.00. Therefore, $B = 1 + Ae^{-2\beta l}\cos(\sigma - 2\beta l)$ approximately, which shows that B may be set equal to 1, and α equal to zero. The resulting solution

$$\frac{\theta}{\theta_o} = e^{-\beta x}\sin(\omega t - \beta x) + Ae^{\beta(x-2l)}\sin[\omega t + \sigma + \beta(x-2l)]$$

will not be in error by more than $e^{-2\beta l}$ in unity.

The problem's main interest lies in the solution of the temperature and heat flow from the inner surface of the wall where $x = l$

$$\text{Here } \frac{\theta}{\theta_o} = \frac{\theta_i}{\theta_o} = e^{-\beta l}[\sin(\omega t - \beta l) + A\sin(\omega t - \beta l + \sigma)]$$

which obviously represents a simple sine function of ampltitude

$$\frac{\theta_m}{\theta_o} = Re^{-\beta l} \quad \textit{where} \qquad (8)$$

$$R = \sqrt{1 + A^2 + 2A\cos\sigma}$$

$\frac{\theta_m}{\theta_o}$ is the ratio of the amplitudes of the temperature variations on the inside of the wall to that on the outside. Since A and σ are functions of the single variable f, R is also a function of f, and is plotted in Fig. 27 for both the fundamental and first harmonic waves. In a like manner, values of $e^{-\beta l}$ are plotted against βl in Fig. 28.

To find the value of $\frac{\theta_m}{\theta_o}$ it is only necessary to calculate f and βl from the constants of the slab, and take the product $Re^{-\beta l}$ from Fig. 27 and Fig. 28. The temperature at the inner surface may now be expressed by the equation

$$\theta_i = \theta_m \sin\omega(t + t_o) \qquad (9)$$

where

t_o = the time between the occurences of the temperature maxima at the outside and inside surfaces. The temperature is maximum at the outside surface when $\omega t = \frac{\pi}{2}$, and the corresponding maximum at the inside surface occurs when

$$\tan\ (\omega t - \beta l) = \frac{1 + A \cos \sigma}{A \sin \sigma} \quad \textit{where} \quad (\omega t - \beta l) \text{ is in the first quadrant.}$$

$$\text{Then } \omega t_o = \left\{ \tan^{-1} \left(\frac{1 + A \cos \sigma}{A \sin \sigma} \right) - \frac{\pi}{2} \right\} + \beta l \qquad (10)$$

After f and βl have been calculated, curves may be drawn for rapid solution of the timing of the waves. Fig. 29 gives the uncorrected time in hours required for the crest of the fundamental and first harmonic waves to pass through a panel. Fig. 30 gives the correction time in hours to be applied in determining the correct length of time required for the fundamental and first harmonic waves to pass through a panel.

Because the English system of units combines terms, which because of their dimensions (*i.e.* feet with inches in Btu per square foot per hour per degree Fahrenheit per inch of thickness) are very difficult to use in involved mathematical processes, all physical constants used in the Laboratory work were converted into the cgs or metric system, and used in that form until all dimensions cancelled in the mathematical process. These constants, being dimensionless, are adaptable to temperature and heat flow measurements in any system of units.

In applying this solution to a practical example, the data collected on the gypsum panel on September 9, 1931, were chosen. Fig. 31 is a graphical analysis of these data. The solid line curve, *BCDEG,* represents the plot of temperatures obtained on the outer surface of the panel. A time for starting was chosen arbitrarily at 4:00 a.m., and the temperatures have been plotted through to 4:00 a.m. on the morning of the following day, September 10, 1931.

To analyze the repeating wave caused by solar radiation on the outer surface of the panel, use is made of the Fourier series, which is a trigonometric series approximation of an arbitrary periodic function, the terms of which are sines and cosines of increasing multiples of the variable. In the solution, the approximate equation of the outer surface temperature curve is found by analysis of the curve. A Fourier series consists of an infinite number of terms; but for ordinary heat flow work a summation of only the first two terms, namely the fundamental and the first harmonic, is sufficient. To determine the constants entering into the series, the temperature curve for the outer surface may be analyzed by the use of either a polar planimeter or a harmonic analyzer, or by computing mathematical averages, which last-named method was used at the Laboratory because it required no special equipment. The two-term mathematical series representing the outer surface temperature was first computed; from this the amplitudes of the fundamental and first harmonic waves at the outer surface were obtained. By applying the theoretical formulae to these amplitudes of the outer surface, the corresponding fundamental and first harmonic amplitudes of the inner surface were obtained. These inner surface amplitudes were vectorially added to give the wave heat flow, which was added algebraically to the

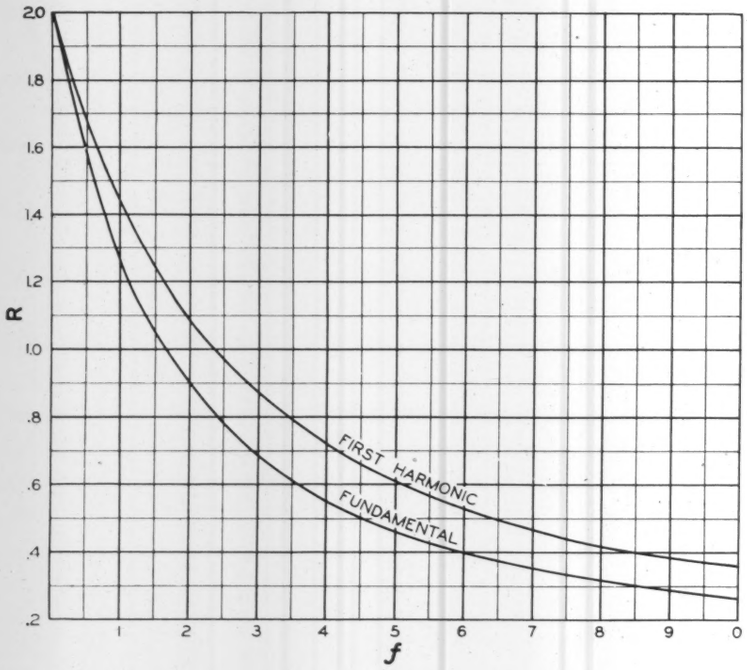


FIG. 27—CURVES GIVING R FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES USED IN CALCULATING THE RATIO OF THE AMPLITUDE OF THE TEMPERATURE OSCILLATION ON THE OUTSIDE WALL SURFACE TO THAT ON THE INSIDE

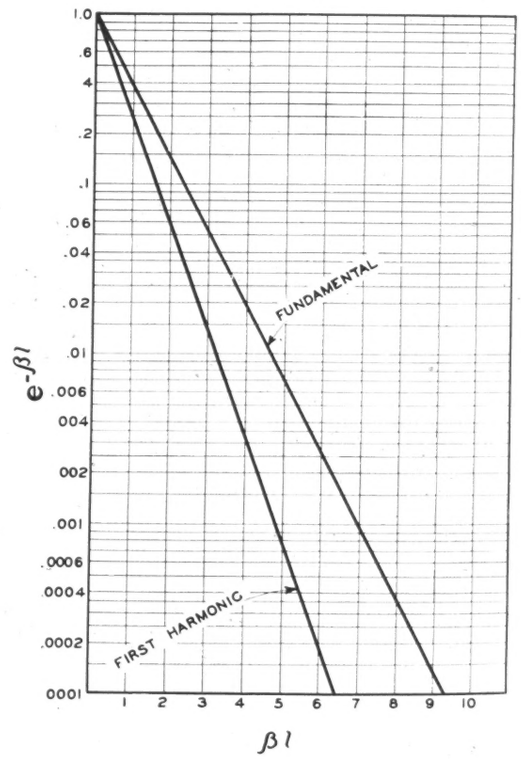


FIG. 28—CURVES GIVING $e^{-\beta l}$ FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES USED IN CALCULATING THE RATIO OF THE AMPLITUDE OF THE TEMPERATURE OSCILLATION ON THE OUTSIDE WALL SURFACE TO THAT ON THE INSIDE

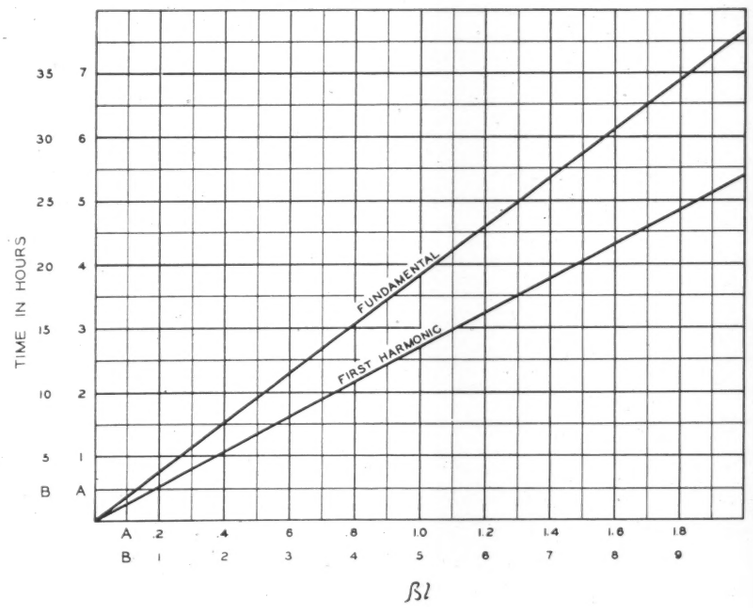


FIG. 29—CURVES GIVING UNCORRECTED TIME IN HOURS REQUIRED FOR THE CREST OF THE FUNDAMENTAL AND FIRST HARMONIC WAVES TO PASS THROUGH A PANEL

heat flow in the panel (considered to be steady) to give the heat flow at the inner surface.

1. On the Fourier Analysis Chart, Table 5, the starting time is filled in at the top of the sheet. The time starts at 4:00 a. m. and continues down the sheet at 20-min increments until 3:40 the following morning has been reached. Thus 72 times have been listed.

2. On each line of the Fourier Analysis Chart, or every twenty minutes during the twenty-four hour period, the value of the surface temperature above zero Fahrenheit is recorded in column *a*. Thus, on Fig. 31 at *N*, which corresponds to 12:20 p. m., the value of the curve is 132.3 F.

3. On the chart, Table 5, the 72 readings in column *a* total 6273.90 F which when divided by 72 gives 87.14 F as the value of the average integrated temperature. This temperature line, *AF*, will divide the curve, Fig. 31, so that the sum of the solid line areas *ABC* and *EFG* will be exactly equal to the area *CDE*. In other words, over a twenty-four hour period, the area above the average integrated temperature line will be exactly equal to the area below it.

4. The value of the average integrated temperature is algebraically subtracted from the temperature listed in column *a* to obtain the values listed in column *b*. These values show the height of the curve *BCDEG* above or below the average integrated temperature line, 87.14 F. Thus, at 1:40 p. m. there is a height of 43.96 F as shown by *P*, while at 11:20 p. m. the height is –19.84 F as shown by *Q*.

5. The values in column *b* are multiplied by the listed values of cos α, sin α, cos 2α and sin 2α to obtain the respective columns *b* cos α, *b* sin α, *b* cos 2α, and *b* sin 2α. All changes in sign in the cosine and sine are taken into account.

6. The 72 readings in each of the columns mentioned in Paragraph 5 are algebraically added. Each of these four totals is divided by 36 to obtain the coefficients for the first two terms, the fundamental and first harmonic of the Fourier series. The series is written:

$$\theta_{\text{outer surface}} = A \cos\alpha + B \sin\alpha + C \cos 2\alpha + D \sin 2\alpha + + + \quad (11)$$

where

A is the coefficient determined from the *b* cos α column

B is the coefficient determined from the *b* sin α column

C is the coefficient determined from the *b* cos 2α column

D is the coefficient determined from the *b* sin 2α column

In Equation 11, the terms containing *A* and *B* will give the fundamental of the series, while the terms containing *C* and *D* give the first harmonic. In the example when the aforementioned operations had been performed on the algebraic totals, the series is found to be:

$$\theta_{\text{outer surface}} = -27.186 \cos\alpha + 20.929 \sin\alpha - 4.308 \cos 2\alpha - 12.900 \sin 2\alpha + + + \quad (12)$$

6a. If a check of the surface temperature is desired, the actual temperature, as measured on the outer surface, may be compared with the Fourier series over a 24-hour period. This check is shown in Table 6. Starting at 4:00 a. m., or zero time, the timing angle α is increased by 15 deg for each hour, since there are 360 deg in a full cycle of 24 hours; the sines and cosines for all of the angles α and 2α are found from trigonometric tables, and the indicated multiplications performed; the results of these multiplications are added algebraically to obtain the amplitude, $\theta_{\text{outer surface}}$; and this resulting temperature wave is added to the value of the average integrated temperature to obtain a temperature which will approximate the experimental temperature curve at the outer surface. The resulting broken line curve, *BCDEG*,

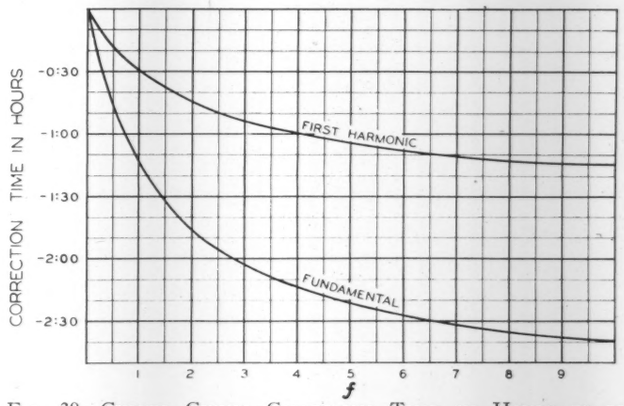


FIG. 30—CURVES GIVING CORRECTION TIME IN HOURS TO BE APPLIED IN DETERMINING THE CORRECT LENGTH OF TIME REQUIRED FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES TO PASS THROUGH A PANEL

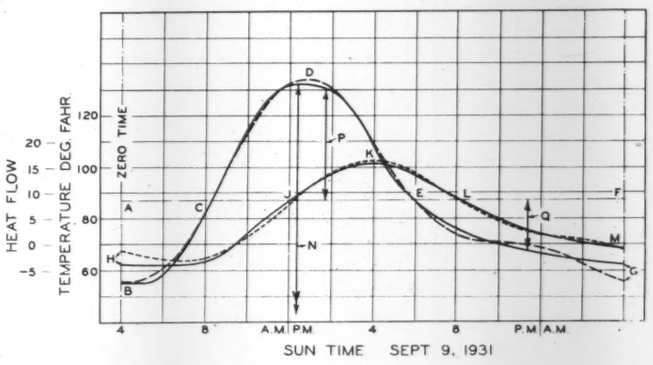


FIG. 31—MATHEMATICAL ANALYSIS OF HEAT FLOW THROUGH 4-IN. GYPSUM PANEL

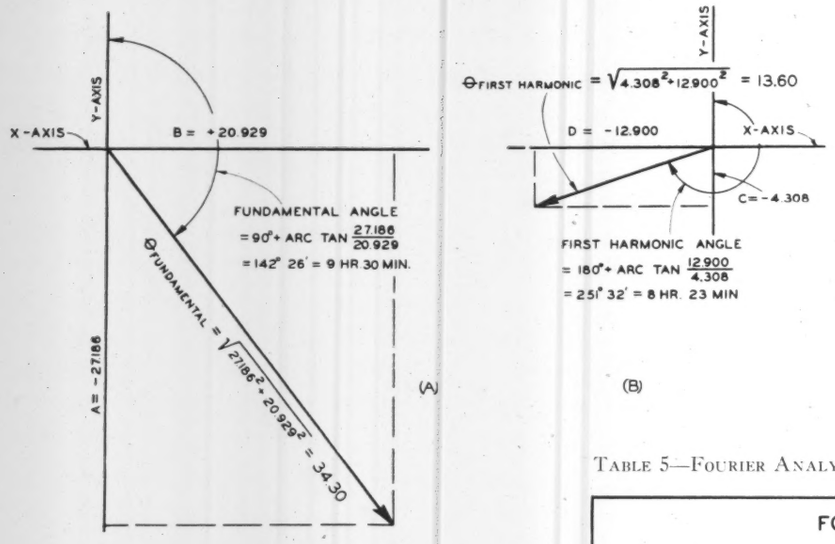


FIG. 32—VECTORIAL DETERMINATION OF FUNDAMENTAL AND FIRST HARMONIC AMPLITUDE AT OUTER SURFACE

in Fig. 31 is the check on the accuracy of the two-term Fourier analysis.

7. From the English or engineering values of the physical constants of the wall, namely

$k = 1.445$ Btu per square foot per hour per degree Fahrenheit per inch

$\rho = 64.89$ pounds per cubic foot

$c = 0.234$

$h_1 = 1.9$ Btu per square foot per hour per degree Fahrenheit

$l = 4.188$ in.

which have been determined previously by test and measurement, the equivalent cgs values are found to be

$k = 1.445 \times 3.44 \times 10^{-4} = 4.971 \times 10^{-4}$ cal per square centimeter per centimeter per second per degree centigrade

$\rho = 64.89 \div 62.4 = 1.04$ grams per cubic centimeter

$c = 0.234$

$h_1 = 1.9 \times 1.355 \times 10^{-4} = 2.574 \times 10^{-4}$ cal per square centimeter per second per degree centigrade

$l = 4.188 \times 2.54 = 10.637$ centimeters

8. The value of β is determined from the formula

$$\beta = \sqrt{\frac{c\rho\omega}{2k}} \qquad (13)$$

where

$$\omega = \frac{2\pi}{T}$$

T = the length of the period. In the cgs system, $T = 86{,}400$ seconds for a 24-hour period.

In the example

$$\beta = \sqrt{\frac{0.234 \times 1.04 \times 3.1416}{86{,}400 \times 4.971 \times 10^{-4}}} = 0.1334$$

9. The value of E is determined from the formula

$$E = \frac{h_1}{k}$$

and in the example

$$E = \frac{2.574 \times 10^{-4}}{4.971 \times 10^{-4}} = 0.5178 \qquad (14)$$

10. The value of f is determined from the formula

$$f = \frac{E}{\beta} \qquad (15)$$

TABLE 5—FOURIER ANALYSIS CHART FOR OUTER SURFACE TEMPERATURE

FOURIER ANALYSIS

GYPSUM PANEL — SEPT. 9, 1931

ZERO TIME 4 A.M. — AV. TEMP. 87.14 F.

Time	a	b	α	Cos α	b Cos α	Sin α	b Sin α	2α	Cos 2α	b Cos 2α	Sin 2α	b Sin 2α
4:00	54.8	-32.34	0	1.000	-32.34	0.000	.00	0	1.000	-32.34	0.000	.00
20	54.8	-32.34	5	.996	-32.21	.087	- 2.81	10	.985	-31.85	.174	- 5.63
40	54.9	-32.24	10	.985	-31.76	.174	- 5.61	20	.940	-30.31	.342	-11.03
5:00	54.9	-32.24	15	.966	-31.14	.259	- 8.35	30	.866	-27.92	.500	-16.12
20	55.0	-32.14	20	.940	-30.21	.342	-10.99	40	.766	-24.62	.642	-20.63
40	56.0	-31.14	25	.906	-28.21	.423	-13.17	50	.642	-19.99	.766	-23.85
6:00	58.0	-29.14	30	.866	-25.24	.500	-14.57	60	.500	-14.57	.866	-25.24
20	60.6	-26.54	35	.819	-21.74	.574	-15.23	70	.342	- 9.08	.940	-24.95
40	64.0	-23.14	40	.766	-17.73	.642	-14.86	80	.174	- 4.03	.985	-22.79
7:00	67.4	-19.74	45	.707	-13.96	.707	-13.96	90	0.000	.00	1.000	-19.74
20	72.0	-15.14	50	.642	- 9.72	.766	-11.60	100	-.174	2.63	.985	-14.91
40	77.2	- 9.94	55	.574	- 5.71	.819	- 8.14	110	-.342	3.40	.940	- 9.34
8:00	82.4	- 4.74	60	.500	- 2.37	.866	- 4.10	120	-.500	2.37	.866	- 4.10
20	87.7	0.56	65	.423	0.24	.906	.51	130	-.642	- .36	.766	0.43
40	93.3	6.16	70	.342	2.11	.940	5.79	140	-.766	- 4.72	.642	3.95
9:00	98.8	11.66	75	.259	3.02	.966	11.26	150	-.866	-10.10	.500	5.83
20	104.0	16.86	80	.174	2.93	.985	16.61	160	-.940	-15.85	.342	5.77
40	109.5	22.36	85	.087	1.95	.996	22.27	170	-.985	-22.02	.174	3.89
10:00	114.6	27.46	90	0.000	.00	1.000	27.46	180	1.000	-27.46	0.000	.00
20	119.0	31.86	95	- .087	- 2.77	.996	31.73	190	-.985	-31.38	-.174	- 5.54
40	123.0	35.86	100	-.174	- 6.24	.985	35.32	200	-.940	-33.71	-.342	-12.26
11:00	126.1	38.96	105	-.259	-10.09	.966	37.64	210	-.866	-33.74	-.500	-19.48
20	128.8	41.66	110	-.342	-14.25	.940	39.16	220	-.766	-31.91	-.642	-26.75
40	130.7	43.56	115	- .423	-18.43	.906	39.47	230	-.642	-27.97	-.766	-33.37
12:00	131.8	44.66	120	- .500	-22.33	.866	38.68	240	.500	-22.33	-.866	-38.63
20	132.3	45.16	125	- .574	-25.92	.819	36.99	250	-.342	-15.44	-.940	-42.45
40	132.4	45.26	130	- .642	- 29.06	.766	34.67	260	-.174	- 7.88	-.985	-44.58
1:00	132.2	45.06	135	-.707	-31.86	.707	31.86	270	0.000	.00	-1.000	-45.06
20	131.9	44.76	140	-.766	-34.29	.642	28.74	280	.174	7.79	- .985	-44.09
40	131.1	43.96	145	-.819	-36.00	.574	25.23	290	.342	15.03	- .940	-41.32
2:00	129.9	42.76	150	- .866	-37.03	.500	21.38	300	.500	21.38	- .866	-37.03
20	128.0	40.86	155	- .906	-37.02	.423	17.28	310	.642	26.23	- .766	-31.30
40	125.6	38.46	160	- .940	-36.15	.342	13.15	320	.766	29.46	- .642	-24.69
3:00	122.3	35.16	165	- .966	-33.96	.259	9.12	330	.866	30.45	-.500	-17.58
20	118.5	31.36	170	- .985	-30.89	.174	5.46	340	.940	29.48	-.342	-10.73
40	114.0	26.86	175	- .996	-26.75	.087	2.34	350	.985	26.46	-.174	- 4.67
4:00	109.4	22.26	180	-1.000	-22.26	0.000	.00	360	1.000	22.26	0.000	.00
20	104.3	17.16	185	- .996	-17.09	-.087	- 1.49	10	.985	16.90	.174	2.99
40	100.0	12.86	190	- .985	-12.67	-.174	- 2.24	20	.940	12.09	.342	4.40
5:00	96.4	9.26	195	- .966	- 8.95	-.259	- 2.40	30	.866	8.02	.500	4.63
20	92.9	5.76	200	- .940	- 5.41	-.342	- 1.97	40	.766	4.41	.642	3.70
40	89.5	2.36	205	- .906	- 2.14	-.423	- 1.00	50	.642	1.52	.766	1.81
6:00	86.7	- 0.44	210	-.866	0.38	-.500	.22	60	.500	- 0.22	.866	- 0.38
20	84.2	- 2.94	215	- .819	2.41	-.574	1.69	70	.342	- 1.01	.940	- 2.76
40	82.0	- 5.14	220	-.766	3.94	-.642	3.30	80	.174	- 0.89	.985	- 5.06
7:00	80.3	- 6.84	225	-.707	4.84	-.707	4.84	90	0.000	.00	1.000	- 6.84
20	79.0	- 8.14	230	-.642	5.23	-.766	6.24	100	-.174	1.42	.985	- 8.02
40	77.4	- 9.74	235	-.574	5.59	-.819	7.98	110	-.342	3.33	.940	- 9.16
8:00	75.9	- 11.24	240	-.500	5.62	-.866	9.73	120	-.500	5.62	.866	- 9.73
20	74.3	- 12.84	245	-.423	5.43	-.906	11.63	130	-.642	8.24	.766	- 9.84
40	73.0	- 14.14	250	-.342	4.84	-.940	13.29	140	-.766	10.83	.642	- 9.08
9:00	72.2	- 14.94	255	-.259	3.87	-.966	14.43	150	-.866	12.94	.500	- 7.47
20	71.3	- 15.84	260	-.174	2.76	-.985	15.60	160	-.940	14.89	.342	- 5.42
40	70.5	- 16.64	265	-.087	1.45	-.996	16.57	170	-.985	16.39	.174	- 2.90
10:00	69.8	- 17.34	270	0.000	.00	-1.000	17.34	180	-1.000	17.34	0.000	.00
20	69.0	- 18.14	275	.087	- 1.58	-.996	18.07	190	-.985	17.87	-.174	3.16
40	68.3	- 18.84	280	.174	- 3.28	-.985	18.56	200	-.940	17.71	-.342	6.44
11:00	67.9	- 19.24	285	.259	- 4.98	-.965	18.59	210	-.866	16.66	-.500	9.62
20	67.3	- 19.84	290	.342	- 6.79	-.940	18.65	220	-.766	15.20	-.642	12.74
40	66.8	- 20.34	295	.423	- 8.60	-.906	18.43	230	-.642	13.06	-.766	15.58
12:00	66.2	- 20.94	300	.500	-10.47	-.866	18.15	240	-.500	10.47	-.866	18.13
20	65.9	- 21.24	305	.574	-12.19	-.819	17.40	250	-.342	7.26	-.940	19.97
40	65.3	- 21.84	310	.642	-14.02	-.766	16.60	260	-.174	3.80	-.985	21.51
1:00	64.9	- 22.24	315	.707	-15.72	-.707	15.72	270	0.000	.00	-1.000	22.24
20	64.4	- 22.74	320	.766	-17.42	-.642	14.60	280	.174	- 3.96	-.985	22.40
40	64.0	- 23.14	325	.819	-18.95	-.574	13.28	290	.342	- 7.91	- .940	21.75
2:00	63.6	- 23.54	330	.866	-20.39	-.500	11.77	300	.500	-11.77	- .866	20.39
20	63.2	- 23.94	335	.906	-21.69	-.423	10.13	310	.642	-15.37	- .766	18.34
40	63.0	- 24.14	340	.940	-22.69	-.342	8.26	320	.766	-18.49	- .642	15.50
3:00	62.8	- 24.34	345	.966	-23.51	-.259	6.30	330	.866	-21.08	- .500	12.17
20	62.5	- 24.64	350	.985	-24.27	-.174	4.29	340	.940	-23.16	-.342	8.43
40	62.2	- 24.94	355	.996	-24.84	-.087	2.17	350	.985	-24.57	-.174	4.34
Totals 627.390					-978.68		+753.44			-155.10		-464.21
Average = 87.14					A = -27.186		B = +20.929			C = -4.308		D = -12.900

TABLE 6—TEMPERATURES RESULTING FROM FOURIER ANALYSIS OF OUTSIDE TEMPERATURE CURVE

OUTER SURFACE TEMPERATURE
GYPSUM PANEL SEPT. 9, 1931

Time	(a) α	(b) Cos α	(c) (b)x A	(d) Sin α	(e) d x B	(f) 2α	(g) Cos 2α	(h) (g)x C	(j) Sin 2α	(k) (j)x D	(m) (c)+(e)+(h)	(n) (m)+Av Temp
4	0	1000	-2719	000	.00	0	1000	-431	.000	.00	-3150	5564
5	15	966	-2626	.259	542	30	866	-373	500	- 645	-3102	5612
6	30	866	-2354	500	1046	60	500	-215	866	-1117	-2640	6074
7	45	707	-1922	707	1480	90	.000	.00	1000	-1290	-1732	6982
8	60	500	-1359	866	1812	120	- 500	215	866	-1117	- 449	8265
9	75	259	- 704	966	2021	150	- 866	373	500	- 645	1045	9759
10	90	000	.00	1000	2093	180	-1000	431	000	.00	2524	11238
11	105	- 259	704	966	2021	210	- 866	373	- 500	645	3743	12457
12	120	- 500	1359	866	1812	240	- 500	215	- 866	1117	4353	13217
1	135	- 707	1922	707	1480	270	.000	.00	-1000	1290	4692	13406
2	150	- 866	2354	500	1046	300	500	- 215	- 866	1117	4302	13016
3	165	- 966	2626	259	542	330	866	-373	- 500	645	3440	12154
4	180	-1000	2719	000	.00	0	1000	-431	000	.00	2288	11002
5	195	- 966	2626	-.259	- 542	30	866	-373	500	- 645	1066	9780
6	210	- 866	2354	-.500	-1046	60	500	-215	866	-1117	- 24	8690
7	225	- 707	1922	-.707	-1480	90	.000	.00	1000	-1290	- 848	7866
8	240	- 500	1359	-.866	-1812	120	- 500	215	866	-1117	-1355	7359
9	255	- 259	704	-.966	-2021	150	- 866	373	500	- 645	-1589	7125
10	270	000	.00	-1000	-2093	180	-1000	431	000	.00	-1662	7052
11	285	259	- 704	-.966	-2021	210	- 866	373	- 500	645	-1707	7007
12	300	500	-1359	-.866	-1812	240	- 500	215	- 866	1117	-1839	6876
1	315	707	-1922	-.707	-1480	270	000	.00	-1000	1290	-2112	6602
2	330	866	-2354	-.500	-1046	300	500	-215	- 866	1117	-2498	6216
3	345	966	-2626	-.259	- 542	330	866	-373	- 500	645	-2896	5818

TABLE 7—CALCULATED HEAT FLOWS THROUGH INNER SURFACE

HEAT FLOW ANALYSIS
GYPSUM PANEL SEPT 9, 1931

Time	(a) Fund Angle	(b) Sin a	(c) (b)x θ1st Harmonic	(d) 1st Har Angle	(e) Sin d	(f) (e)x θ1st Harmonic	(g) (c)+(f)	(j) (g)x h_i	(k) Ah steady Flow
4	259°15	- 982	-4.55	1350°	.920	1.27	-3.28	-6.23	- 110
5	274°-	- 997	-4.61	143°-	.620	83	-378	-718	- 205
6	289°-	- 944	-4.37	173°-	.122	.17	-4.20	-798	- 285
7	304°-	- 827	- 383	203°-	-.391	- .54	-4.37	-830	- 317
8	319°-	- 653	- 302	233°-	- 798	-1.10	-4.12	-783	- 270
9	334°-	- 434	- 200	263°-	- 993	-1.37	-3.37	-640	-1.27
10	349°-	- 186	- .86	293°-	- 920	-1.27	-2.13	-4.04	1.09
11	4°15	.074	.34	323°-	- 602	- .83	- .49	- .93	420
12	19°-	.330	1.53	353°-	- 122	- .17	1.36	258	771
1	34°-	.563	2.60	23°0'	391	.54	3.14	597	11.10
2	49°-	.757	3.50	53°-	798	1.10	4.60	874	1387
3	64°-	.901	4.17	83°-	993	1.37	5.54	1052	1565
4	79°-	.982	4.55	113°-	920	1.27	5.82	11.06	1619
5	94°-	.997	461	143°-	602	.83	5.44	1034	1547
6	109°-	.944	4.37	173°-	122	.17	4.54	863	1376
7	124°-	.827	3.83	203°-	-.391	- .54	3.29	625	11.38
8	139°-	.653	3.02	233°-	-.798	-1.10	1.92	365	878
9	154°-	.434	2.00	263°-	- 993	-1.37	.63	1.19	632
10	169°-	.186	.86	293°-	- 920	-1.27	- .41	- .78	435
11	184°-	-.074	- .34	323°-	- 602	- .83	-1.17	-2.22	291
12	199°-	-.330	-1.53	353°-	- 122	- .17	-1.70	-3.23	190
1	214°-	-.563	-2.60	23°-	.391	.54	-2.06	-3.91	1.22
2	229°-	-.757	-3.50	53°-	798	1.10	-2.40	-4.56	.57
3	244°-	-.901	-4.17	83°-	993	1.37	-2.80	-5.32	- .19

and in the example

$$f = \frac{0.5178}{0.1334} = 3.881$$

11. The value of βl is determined from the formula

$$\beta l = \beta \times l \tag{16}$$

and in the example

$$\beta l = 0.1334 \times 10.637 = 1.419$$

12. Next, the value of the fundamental amplitude at the outer surface and the time in hours that its crest is distant from zero time are found by analyzing the fundamental—or A and B—terms of the Fourier equation of the wave form at the outer surface, by drawing the vector diagram as shown in A in Fig. 32. Here, A is laid off on the y axis taking into account its sign. In the example, —27.186 is the A term, and this is laid off on the y axis in a downward direction from the origin. The B term is laid off on the x axis also taking into account its sign. In the example, 20.929 is the B term, and this is laid off on the x axis to the right. The resultant, which is the value of the fundamental amplitude at the outer surface is therefore

$$\theta_{\substack{\text{fundamental}\\\text{outer surface}}} = \sqrt{27.186^2 + 20.929^2} = 34.30$$

and the timing is the angular distance from the positive y axis measured in a clockwise direction, or

$$90 \text{ deg} + \text{arc tan } \frac{27.186}{20.929} = 142 \text{ deg } 26 \text{ min.}$$

Expressed in hours, this is 9 hr 30 min, and is the time it takes the crest of the fundamental wave to travel from zero time to the time of maximum crest at the outer surface. This is found by dividing the angular distance by 15 deg, the number of degrees in one hour of the fundamental.

13. The value of the first harmonic amplitude at the outer surface is next determined, as is also the time in hours that its crest is distant from zero time. These are found by vector analysis exactly as were those of the previous determination. As shown in B of Fig. 32, the constant in the Fourier equation of the wave form at the outer surface C, or –4.308, is laid off on the y axis in a downward direction. The constant D, –12.900, in the same equation, is laid off on the x axis to the left. The resultant is the value of the amplitude of the first harmonic at the outer surface

$$\theta_{\substack{\text{first harmonic}\\\text{outer surface}}} = \sqrt{4.308^2 + 12.900^2} = 13.60$$

Again, the timing is the angular distance from the y axis measured in a clockwise direction, or

$$180 \text{ deg} + \text{arc tan } \frac{12.900}{4.308} = 251 \text{ deg } 32 \text{ min}$$

Expressed in hours, it takes the crest of the first harmonic 8 hr 23 min, to travel from zero time to the time of maximum crest at the outer surface, since for the harmonic, there are 30 deg in one hour.

14. From the value of f and the curves in Fig 27, R is determined to be 0.560 for the fundamental and 0.740 for the first harmonic wave.

15. From the value of βl and the curves in Fig. 28, $e^{-\beta l}$ is determined to be 0.241 for the fundamental and 0.137 for the first harmonic wave.

16. The value of the amplitude of the fundamental at the inner surface is determined from the multiplication of the amplitude of the fundamental at the outer surface by the factors R and $e^{-\beta l}$ for the fundamental. In the example,

$$\theta_{\substack{\text{fundamental}\\\text{inner surface}}} = 34.30 \times 0.560 \times 0.241 = 4.629$$

17. The value of the amplitude of the first harmonic at the inner surface is determined from the multiplication of the amplitude of the first harmonic at the outer surface by the factors R and $e^{-\beta l}$ for the first harmonic. In the example,

$$\theta_{\substack{\text{first harmonic}\\\text{inner surface}}} = 13.60 \times 0.740 \times 0.137 = 1.379.$$

18. Next, the uncorrected time in hours and minutes it takes the crest of the fundamental wave to travel through the panel is determined. On the fundamental curve in Fig. 29, the time is read from the value of βl to be 5 hr 25 min. From this is subtracted the corrective time, 2 hr 12 min read from f on the fundamental curve in Fig. 30. This 3 hr 13 min is the true time it takes the crest of the fundamental wave to pass through the finite wall.

19. The uncorrected time in hours and minutes it takes the crest of the first harmonic wave to travel through the panel is determined. On the first harmonic curve in Fig. 29, the time is read from the value of βl to be 3 hr 50 min. From this is subtracted the corrective time, 59 min, read from f on the first harmonic curve in Fig. 30. This 2 hr 51 min is the true time it takes the crest of the first harmonic wave to pass from the outside to the inside of the finite wall.

20. The time the crest of the fundamental wave is distant from twelve midnight of the night previous is determined by

adding together the time from midnight to zero time, the time from zero time to the time of maximum crest of the fundamental, and the corrected time it takes the wave to pass through the wall. This is done in the example:

4 hr	00 min
9 hr	30 min
3 hr	13 min
16 hr	43 min

Therefore, at 4:43 p. m. there is a crest of the fundamental at the inner surface.

21. The time the crest of the first harmonic wave is distant from twelve midnight of the night previous is determined by adding together the time from midnight to zero time, the time from zero time to the time of maximum crest of the first harmonic, and the corrected time it take the first harmonic to pass through the wall. This is done in the example:

4 hr	00 min
8 hr	23 min
2 hr	51 min
15 hr	14 min

Therefore, at 3:14 p. m. there is a crest of the first harmonic at the inner surface.

22. For ease of computation, it is desirable to determine, from the time of the crest of the fundamental wave at the inner surface, when the timing angle of the fundamental wave will be equal to zero degrees. To obtain this time, it is only necessary to subtract six hours from the time of the crest—the equivalent of subtracting 90 deg from the timing angle. Since 15 deg are equal to one hour for the fundamental wave, the angle of the fundamental at each hour of the day is easily found by interpolation. In the example:

4:43 p. m. –6 hr = 10:43 a. m. at zero angle.
At 11:00 a. m., the angle of the fundamental

$$= \frac{17 \times 15}{60} = 4 \text{ deg } 15 \text{ min}$$

23. In a manner similar to that described, three hours are subtracted from the time of the crest of the first harmonic wave at the inner surface to obtain the time when the timing angle of the first harmonic wave will be equal to zero degrees. Since 30 deg are equal to one hour for the first harmonic wave, the angle of the first harmonic at each hour of the day is easily found by interpolation. In the example:

3:14 p. m. –3 hr = 12:14 p. m. at zero angle.
At 1:00 p. m., the angle of the first harmonic

$$= \frac{46 \times 30}{60} = 23 \text{ deg } 0 \text{ min}$$

24. The average integrated temperature at the inner surface over the twenty-four hour period is determined in the English system of units

$$\frac{(T_1 - T_2)\ k}{l} = (T_2 - T_3)\ h_1 \qquad (17)$$

where

T_1 = average integrated outer surface temperature
T_2 = inside surface temperature
T_3 = inside air temperature

In the example:

$$\frac{87.14 - T_2}{4.188} = (T_2 - 69.6)\ 1.9$$

$(T_2 = 72.30$ F.)

25. The steady heat flow through the inner surface is easily determined from the film transmittance coefficient.

$$\text{Btu per square foot per hour} = (T_2 - T_3)\ h_1 \qquad (18)$$

and in the example is equal to $(72.30 - 69.6)\ 1.9 = 5.13$ Btu

26. The vector analysis form chart, Table 7, is filled out for the two term analysis. The angles of the fundamental waves *a* and first harmonic waves *d* are spaced out, keeping in mind that 15 deg is the equivalent of one hour for the fundamental wave and 30 deg is the equivalent of one hour for the first harmonic wave. The corresponding sines *b* and *e* are multiplied by $\theta_{\text{inner surface}}^{\text{fundamental}}$ or 4.629, and $\theta_{\text{inner surface}}^{\text{first harmonic}}$ or 1.379, as shown in *c* and *f* respectively.

27. The items, *c* and *f*, are algebraically added to obtain the resulting temperature at the inner surface above and below the average surface temperature as shown in *g*.

Multiplying these values by the film transmittance coefficient, h_1, gives the amplitude of the heat flow wave in *j*, and the value of the steady heat flow added to this wave flow results in the calculated heat flow for the wall in *k*. This calculated curve is sketched in Fig. 31 as the broken line curve, *HJKLM*, which checks the solid line observed heat flow curve *HJKLM*.

The foregoing analysis has been applied to a number of tests made on the panels studied. The inner surface heat flow curves marked *calculated* in Figs. 13 to 23 show how closely the results of the application of the mathematical study fit the measured heat flows through the inner surfaces of the panels.

The calculated heat flow through the lower surface of the pine panel, Fig. 13, fits the observed heat flow almost perfectly in regard to both the amplitude and the time of the crest of the wave. There is usually a small break in the calculated curves at 4 a.m., when the arbitrarily chosen, or zero time, begins and ends the daily cycle. The average integrated temperatures of the outer surface for two consecutive days were never quite the same, and since each 24-hour period was calculated as a unit, the adjacent ends of the calculated curves do not abut.

The check between the calculated and observed heat flows through the inner surface of the 6-in. concrete panel, Fig. 14, is not quite as perfect, but considering the greater thickness, conductivity, and heat capacity of this panel, the check is satisfactory. The 4-in. gypsum panel, Fig. 17, gives as close a check as could be desired. The data for September 8 were included for the special purpose of showing the application of the mathematical solution to a day which was far from perfect, and for which the top surface temperature curve deviated greatly from the general type. This day was intermittently and rather densely cloudy from 10 a.m. until 3 p.m. The top surface temperature curve is similar in shape to the observed heat flow curve, and the Fourier analysis of the temperature curve gives a calculated heat flow curve which fits the observed heat flow curve very well.

The checks between the calculated and observed heat flows through the inner surface of the cork-concrete and concrete-cork panels, Figs. 18 and 19, are not as perfect as for the pine and gypsum. The calculated curve for the cork-concrete panel shows about the same amplitude as that for the observed, but its crest is shown to pass through the lower surface about 1½ hours later. The calculated curve for this panel, when the concrete was up, shows a lower amplitude than the observed curve, and lags approximately 3½ hours behind it. The application of the Fourier analysis to the cork-concrete and concrete-cork panels involves the use of the average conductivity, density, and specific heat of the compound panel, so it obviously should not rigorously apply. The curves of Figs. 18 and 19 indicate that the check between observed and calculated heat flow is fair when the insulation is on the outer surface, but unsatisfactory when the insulation is on the inside. These tests indicate that for compound walls, where the physical prop-

erties of the component parts differ widely, an adaptation of the Fourier analysis method which considers each component part separately is desirable.

The calculated heat flow does not fit the observed heat flow perfectly for the lower surface of the laminated 2-in. and 4-in. iron-cork panels, Figs. 20 and 21. No doubt, this is due in part to the fact that these panels had very high heat capacity. It was also more difficult to evaluate accurately the density and specific heat of the cork laminations which were gasket cork and not the variety of cork board used extensively for insulation. Another difficult factor to determine for these panels was the contact resistance between laminations. While the laminations were held firmly together by the weight of the iron for the 4-in. and 8-in. panels, and by bolts outside the center 2-ft square test section for the 2-in. panel, no doubt air films were figured in as cork when calculating density and specific heat. The effect that large air spaces would have on a wall is also a problem with a much more difficult solution.

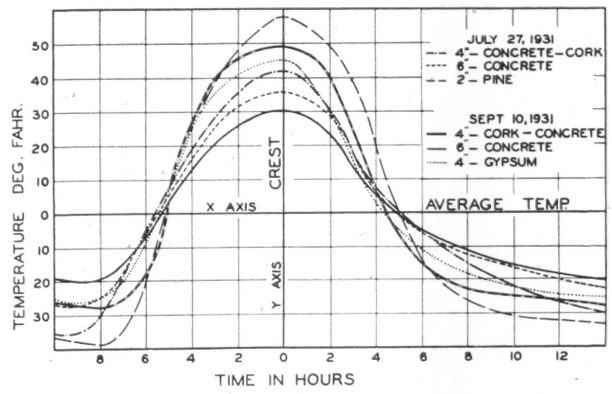


FIG. 33—CURVES FOR UPPER SURFACE TEMPERATURE FOR SEVERAL PANELS SUPERIMPOSED WITH THEIR CRESTS ON THE y AXIS AND THEIR AVERAGE TEMPERATURES ON THE x AXIS

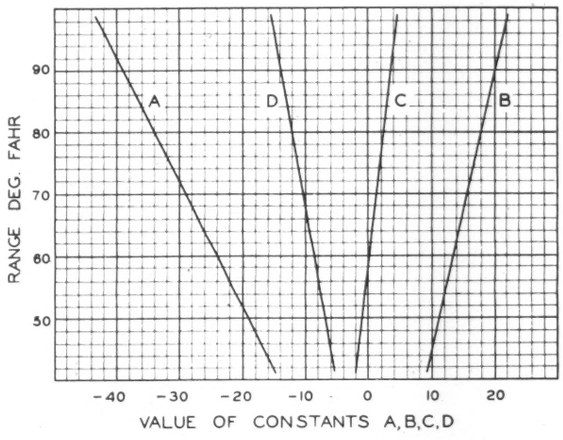


FIG. 34—CURVES GIVING CONSTANTS A, B, C, AND D IN THE FOURIER EQUATION FOR EMPIRICAL DETERMINATION

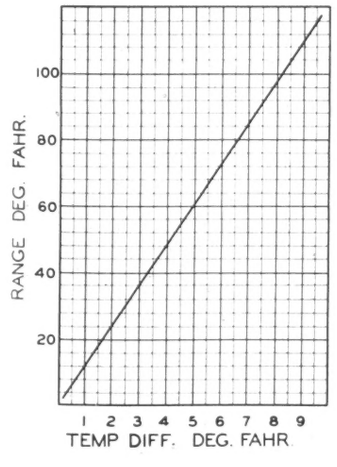


FIG. 35—CURVE GIVING DIFFERENCE BETWEEN THE MEAN OF THE MAXIMUM AND MINIMUM SURFACE TEMPERATURES AND THE AVERAGE INTEGRATED TEMPERATURE FOR EMPIRICAL DETERMINATION

To give additional information on the error introduced by averaging the physical properties of a compound wall, a few days' data were analyzed from the curve for the top surface temperature in contact with the outside air and the curve for the temperature under the upper heat flow meter. The results of this study are shown in Figs. 22 and 23 for pine and concrete. There was little difference in temperature between the top surface and the surface under the meter for the pine panel, and consequently little difference between the calculated heat flow through the lower surface resulting from these two calculations. The curve calculated from the upper surface temperature is not drawn as it falls practically on the curve calculated from the temperature under the meter. Curves for concrete, resulting from both calculations, are drawn in Fig. 23. In this case, there is considerable difference in temperature between the top surface and the surface under the meter. However, the curves for heat flow as calculated from these two temperature curves check very accurately. The check of the two calculations for gypsum was approximately the same as that for pine and is not shown. Figs. 22 and 23 also include the Fourier analysis curves for the upper surface temperature, curves for the outer air temperature in the shade, and curves of the observed inner surface temperature. It is of interest to note that the temperature of the outside air, observed at a level 12 in. above but in the shade at one side of the top surface of the panel, is considerably below the temperature of the top surface.

Empirical Solution of Heat Flow Using Limited Data

The rigid mathematical solution of the periodic heat flow problem requires rather laborious and extensive computations to be made on the outside surface temperature curve. It was desirable to decrease the amount of data required for the solution, as well as the labor of making the solution. The similarity of all the top surface temperature curves for the roof panels as drawn in Figs. 13 to 23, suggested the possibility of setting up a Fourier series directly from the temperature range between the crest and trough, or the maximum and minimum temperatures, of a given surface for the 24-hour period.

Six outside surface temperature curves for as many different roof panels on two different days are all plotted in Fig. 33. The time of the crest of each curve is plotted as zero time, and the average integrated temperature for the 24-hour period as zero temperature. The superimposed curves give a somewhat consistent series, the characteristics of which are rather accurate functions of the amplitude or temperature range. It should follow that the constants in the equation resulting from a

Fourier analysis of these curves should be approximate functions of the range. The Fourier equation is

$$\theta_{\text{outer surface}} = A \cos \alpha + B \sin \alpha + C \cos 2\alpha + D \sin 2\alpha + + + \quad (19)$$

where

A, B, C, D are constants to be determined, and α is the timing angle.

Plotting the six values of each of the constants, A, B, C, and D, as found by analysis of the six curves, Fig. 33, against the range for the respective curves, the points fall fairly close to the curves for these constants in Fig. 34; and hence, to the same degree of accuracy, the constants, for any equation for any day or any roof, within the limits of those included in Fig. 33 may be taken from these curves.

Likewise, the average integrated temperature of the top surface for a 24-hour period may be found by subtracting the temperature difference for the proper range in Fig. 35 from the mean between the maximum and minimum of the outside surface temperatures for the day. In Fig. 24, the use of the empirical method for a determination of an approximation of the Fourier series is illustrated for the 2-in. pine panel on July 28. The crest temperature is 147.3 F, the trough temperature is 59.8 F, the range is 87.5 F, and the average between the crest and trough is 103.6 F. From Fig. 34, the values of the constants A, B, C, and D are therefore: -37.70, $+19.4$, $+3.5$, and -13.8 respectively. From Fig. 35, the difference between the mean of the maximum and minimum temperatures and the average integrated temperature is found to be 7.8 F; and the average integrated temperature is $103.6 - 7.8 = 95.6$ F. The equation for the top surface temperature is

$$\theta_{\text{outer surface}} = -37.7 \cos \alpha + 19.4 \sin \alpha + 3.5 \cos 2\alpha - 13.8 \sin 2\alpha + + +$$

After this temperature equation has been determined, a solution of the heat flow is obtained by continuing from Item 7 of the mathematical solution.

The calculated heat flow through the inner surface, based upon the empirical method, is shown in Figs. 24, 25, and 26 for the pine, concrete, and gypsum panels. These solutions are very satisfactory, and indicate that the method is adaptable for any roof panel within the range of those considered in Fig. 33.

Applications to Other Than Horizontal Roofs of Simple Homogeneous Construction

The data collected and analyzed are for horizontal surfaces only. However, since the primary data considered the outside surface temperature, the methods of calculation presented will apply equally well to structures in any position, as long as the analysis is made of outside surface temperatures.

The curves in Fig. 36 give, *first*, the angles between the direction of the sun's rays and either a horizontal surface, an east, a south, or a west wall; and, *second*, the azimuth angle, or the angle between the direction of the sun's rays and the north. The angles were calculated from data obtained from the American Ephemeris and Nautical Almanac.

The intensity of solar radiation perpendicular to the direction of the sun's rays as observed by the Laboratory pyrheliometer is given in Fig. 37, which represents the maximum intensity of solar radiation observed at the A. S. H. V. E. Laboratory during the three years' study of the subject, and was actually observed on September 7, 1931. In making up the similar curves for July 1 and August 1, in Figs. 38 and 39, the same intensity curve was used by broadening it out for the longer period of sunshine on these days. These curves, therefore, give the maximum solar radiation which may be expected for these months on very clear days with perfect sunshine. Figs. 37 to 39 also show the intensity of solar radiation on a horizontal surface, and on vertical surfaces facing east, south, and west for sun time at Pittsburgh, 40 deg latitude and 80 deg longitude. The intensities on surfaces other than those perpendicular to the sun's rays were calculated from the formula

$$H\phi = H_p \sin \phi \quad (20)$$

where

$H\phi$ = the intensity of radiation in Btu per hour per square foot on the surface whose angle with the direction of the sun's rays is ϕ.

H_p = the intensity of radiation normal to the direction of the sun's rays, or that observed by the pyrheliometer.

It will be noted that the intensity of radiation on any surface not normal to the sun's rays varies with the month. It should be emphasized that the curves in Figs. 36 to 39, other than those showing the intensity of solar radiation for a surface normal to the sun's rays, will vary with the latitude, and if standard time is used, they will also vary by ± one-half hour, with longitude east and west

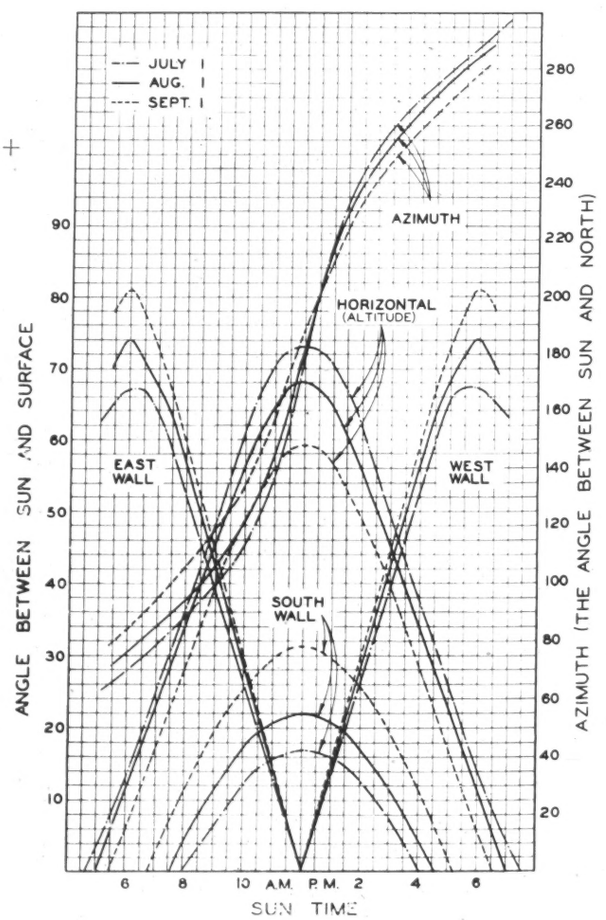

JULY 1
AUG. 1
SEPT. 1
AZIMUTH
HORIZONTAL (ALTITUDE)
EAST WALL
WEST WALL
SOUTH WALL
ANGLE BETWEEN SUN AND SURFACE
AZIMUTH (THE ANGLE BETWEEN SUN AND NORTH)
SUN TIME
6 8 10 A.M. P.M. 2 4 6
10 20 30 40 50 60 70 80 90
20 40 60 80 100 120 140 160 180 200 220 240 260 280

Fig. 36—Curves Giving Azimuth and the Angle Between the Sun and a Horizontal Surface and Wall

of the center of the Standard Time Belt. Similar data, collected on the Laboratory recording pyrheliometer, given in Figs. 15 and 16, indicate the variation in solar intensity from day to day.

The circle points in Fig. 37 were taken from the chart of the recording pyrheliometer of the U. S. Weather Bureau as observed on September 7, 1931, at the Pittsburgh Weather Station, approximately five miles from the A. S. H. V. E. Laboratory. The Weather Bureau instrument is unlike either the Smithsonian silver disk instrument or the Laboratory instrument, because it records the intensity of solar radiation on a horizontal surface. Hence, the points from the Weather Bureau record correspond to the curve calculated from the

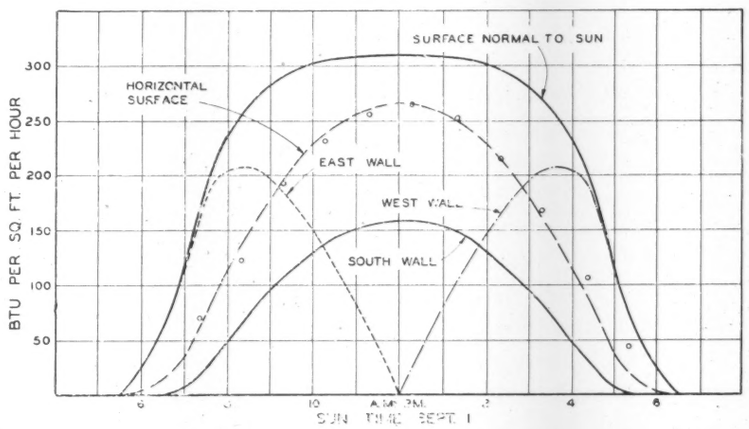


FIG. 37—CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE, AND ON WALLS FOR SEPTEMBER 1. POINTS GIVE WEATHER BUREAU DATA FOR HORIZONTAL SURFACE

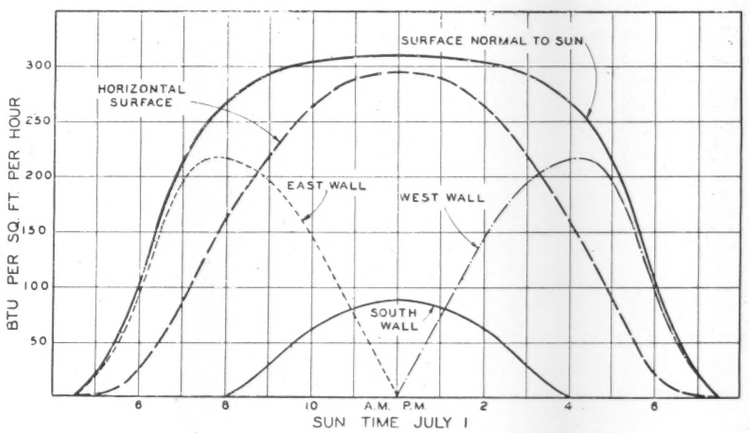


FIG. 38—CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR JULY 1

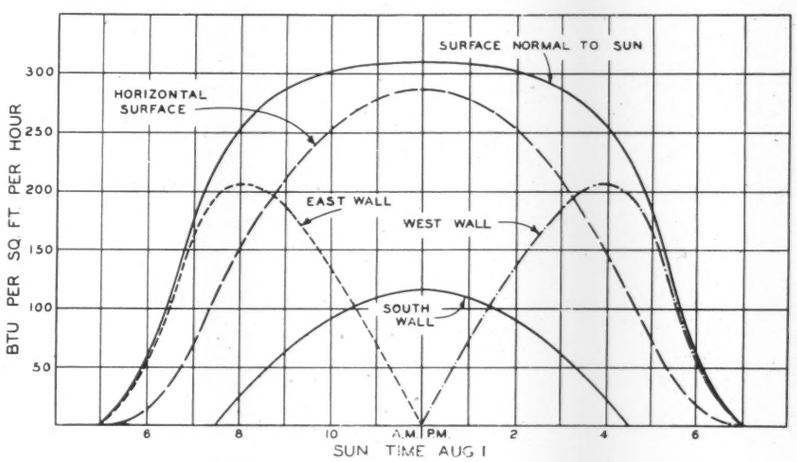


FIG. 39—CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

Laboratory pyrheliometer for a horizontal surface, which curve they fit almost exactly. This is an interesting check for two instruments of widely different design, calibrated entirely independently, and located at stations five miles apart in a city.

The main object of the study being to obtain data for typical hot summer days, favorable weather conditions had to be anticipated by careful interpretation of Weather Bureau reports; a very light haze, or a few scattered clouds would make a test erratic. Many data were obtained which had to be discarded because, after several hours of good weather, clouds blanketed the sun, or a thunderstorm suddenly came up and wet the surfaces.

An attempt was made to choose test panels which would cover a wide range of typical building construction, and which could be used in the application of the theory. Panels used were either homogeneous or constructed of two homogeneous materials. It was decided not to confuse the work with complex types of materials involving air spaces and non-homogeneous materials which would require a more complicated mathematical solution. The study could well be continued on such complex materials, which would require consideration of confusing factors such as convection currents and additional film resistance coefficients. The determination of proper physical constants for such calculations would be very difficult, and many disappointing results should be anticipated. The method given in this paper will give a satisfactory solution for such complex structures if physical constants which will fit the conditions can be assumed for the non-homogeneous panel.

It should be emphasized that the Laboratory's work dealt with roofs and walls without giving consideration to window spaces. Earlier Laboratory tests[8] have shown that on July 1 ordinary single-strength window glass reduces the sun's intensity by about ten per cent, and allows ninety per cent to pass through and become effective as heat in the interior of the room. The data in Fig. 7 show that a glass surface normal to the sun allows about 275 Btu per square foot per hour to pass through it. Accepting this value, calculation shows that a horizontal glass window would allow 260 Btu to pass through it, and that an east and west vertical window would pass 196 Btu, and a south vertical window, 81 Btu. Studies of the effects had by window shades of different reflecting powers placed inside or outside the glass would be of interest and possible value.

The color of the surface is important. As has been pointed out in an earlier Laboratory report,[8] a dead black body which will absorb approximately 100 per cent of the solar radiation will absorb only 70 per cent when covered with a type of red paint, and only 30 per cent when aluminum paint is used. This point is worthy of further consideration as a possible means of reducing heat absorption.

The application of the commonly accepted surface resistance coefficients to the calculation of heat flow is not satisfactory when the sun shines on the outer surface of a wall. As shown in the data, the air temperatures in the shade may be so much lower than the surface temperatures on a hot sunny day because of solar radiation that the heat flow would actually be positive, although the surface coefficients as ordinarily used might indicate a negative heat flow.

[8] See Bibliography, *d*

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Chemical Composition of Coal

The question of the chemical composition of coal is of both scientific and practical interest. A knowledge of the chemical constituents in coal would help in determining its origin and aid in its more efficient utilization, points out the United States Bureau of Mines, Department of Commerce.

The class of compounds which make up the major part of coal is known as the humins. These compounds are considered to be the degradation products of lignin or cellulose, or both. They are complex organic substances of high molecular weight whose structure is little known. However, it is usually assumed that they contain a large number of benzine nuclei linked together in various ways. Recent evidence obtained in the Coal Constitution Laboratory at the Pittsburgh Experiment Station of the Bureau of Mines, while not conclusive, does not bear this out. It appears that most of the nuclei are probably of a heterocyclic nature containing oxygen as well as carbon.

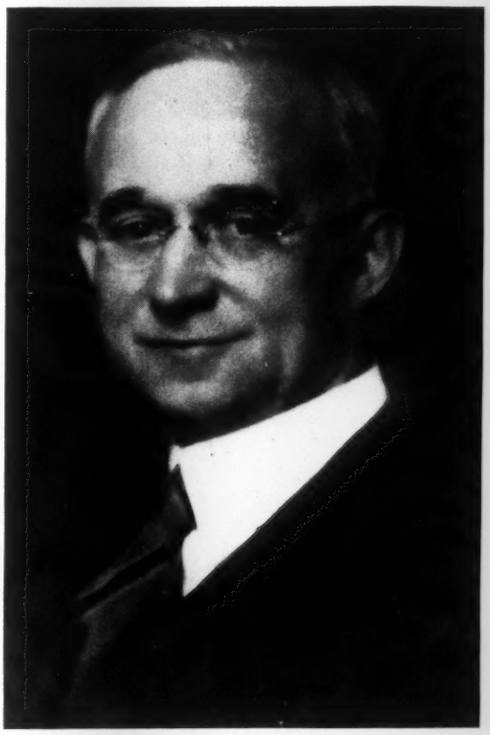
Work in this laboratory with these humic compounds has emphasized the fact that their colloidal nature must be constantly kept in mind when interpreting their reactions. As negatively charged lyophilic colloids, they readily absorb positive ions, and in so doing undergo changes which may appear to be chemical while they are only physical.

It is known that drying of humic compounds greatly reduces their reactivity. This is generally attributed to polymerization or intramolecular condensation of some kind. It seems probable from evidence obtained that much of this inactivation is simply due to irreversible coagulation of these colloidal substances.

Officers and Council Members

F. B. ROWLEY
Minneapolis, Minn.
President

W. T. JONES
Boston, Mass.
First Vice-President



C. V. HAYNES
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Second Vice-President

F. D. Mensing
Philadelphia, Pa.
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W. H. Carrier
Newark, N. J.
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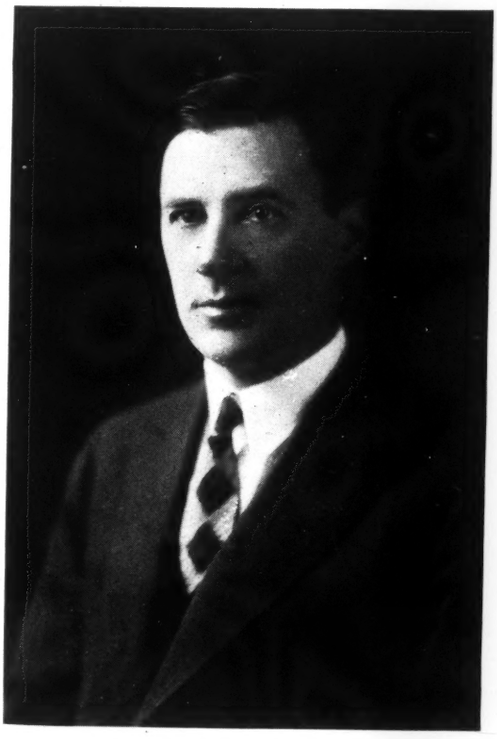
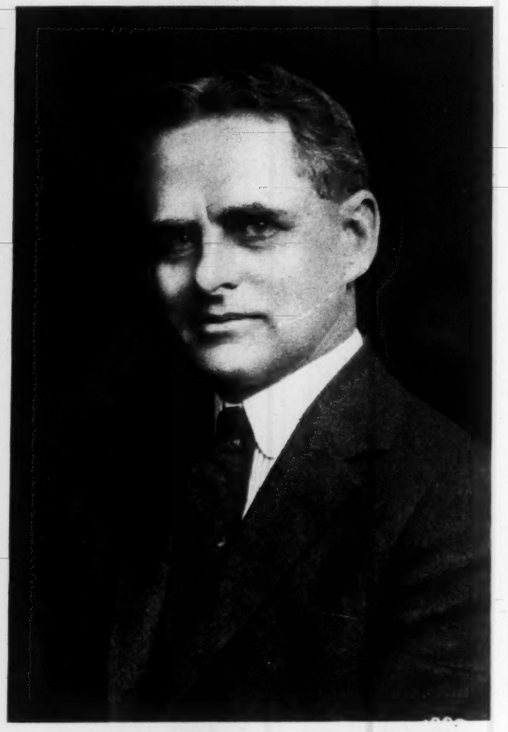


J. F. McIntire
Detroit, Mich.

W. E. Stark
Cleveland, Ohio

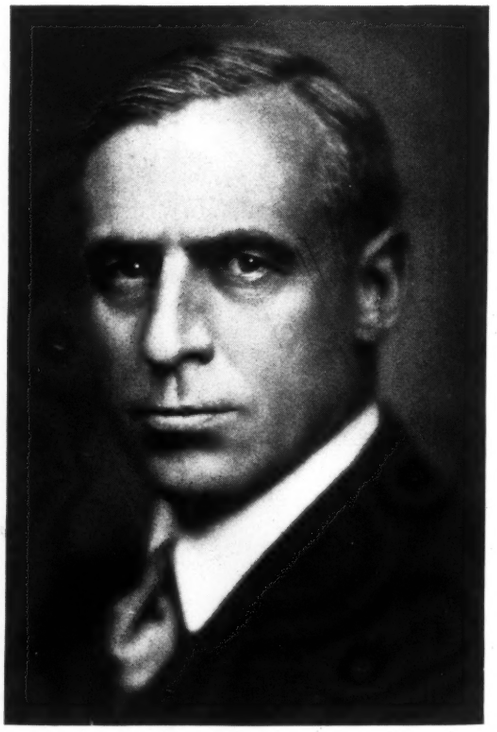


E. K. CAMPBELL
Kansas City, Mo.

E. O. EASTWOOD
Seattle, Wash.

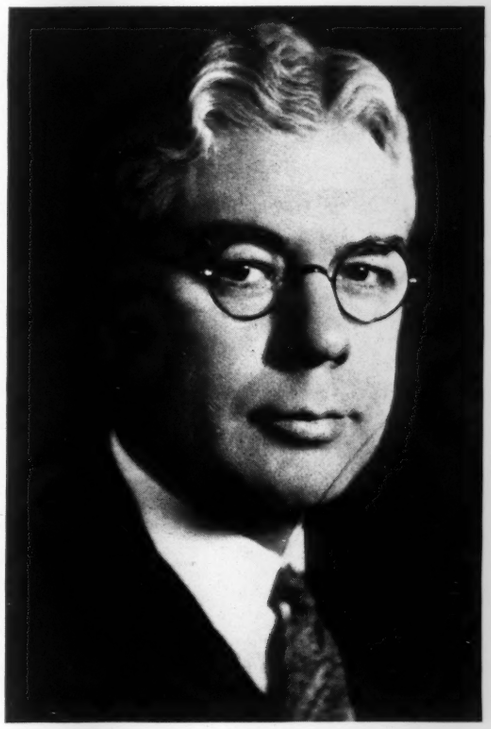
Members of the Council for Two Years

ROSWELL FARNHAM
Buffalo, N. Y.



E. HOLT GURNEY
Toronto, Ont.

D. S. Boyden
Boston, Mass.



R. H. Carpenter
New York, N. Y.

Members of the Council for One Year

J. D. Cassell
Philadelphia, Pa.

John Howatt
Chicago, Ill.

SKYLINE OF MILWAUKEE, WIS.

Milwaukee Invites You

FROM June 27 to 29, members of the Society will assemble in Milwaukee, Wis., at the Hotel Pfister, for the Semi-Annual Meeting, 1932, where they will be the guests of the Wisconsin Chapter of the A. S. H. V. E.

John S. Jung is General Chairman of Arrangements and he will be assisted by the following committee chairmen: Advisory, H. W. Ellis; Reception and Registration, G. L. Larson; Transportation, A. M. Wagner; Finance, V. A. Berghoefer; Entertainment, H. F. Haupt; Golf, E. A. Jones; Banquet, F. G. Weimer; Publicity, Ernest Szekely; Ladies' Entertainment, C. H. Randolph and Mrs. E. A. Jones, Hostess.

It is planned to hold Technical Sessions from 9:30 to 12:30 each morning in the ballroom of the Hotel Pfister. After the sessions, members will have an opportunity to play golf and take part in other forms of recreation which have been planned by the Milwaukee members. A Council Meeting has been scheduled for Monday night and the Semi-Annual Banquet and Dance will be held on Tuesday evening, June 28.

It is expected that some very interesting papers will be presented and reports will be made by the special and technical committees of the Society, including the Committee on Ventilation Standards, on the progress which they have made since the January meeting in Cleveland.

In the very near future a letter of invitation from the Wisconsin Chapter will be received by members of the Society and this will be followed by a circular describing the City of Milwaukee, a program, railroad certificate, and hotel reservation card.

Milwaukee, in years gone by, was an Indian village famed for its climate and natural scenery. It is now one of the leading industrial centers of the country, but still retains many of its early attractions and a great deal of natural beauty. Its park and boulevard systems are a source of pride among its own citizens and are always admired by visitors. Situated on Lake Michigan, an opportunity is offered for bathing and yachting. Public and private golf courses are conveniently located and readily accessible.

Milwaukee is the gateway to the famous Land O' Lakes district in Wisconsin. In this region one sparkling lake after another greets the tourist who motors through mile after mile of virgin timberlands. Unique camps scattered throughout the state of Wisconsin afford fishing, hunting and other outdoor sports, while palatial resorts on shores of Wisconsin lakes offer comfort and luxury to their guests. Thus Milwaukee is an ideal starting point for a summer vacation.

The members of the Wisconsin Chapter are looking forward to a large attendance and are working up an attractive program for the visiting members.

NEW PFISTER HOTEL

Case Conference on Air Conditioning

SEVERAL hundred scientists, engineers, industrialists and students from 12 states and 2 Canadian provinces participated in the Conference on Air Conditioning, held at the Case School of Applied Science, Cleveland, Ohio, March 17-19, 1932. The conference of six sessions started Thursday afternoon under the chairmanship of Prof. F. H. Vose, head of the Mechanical Engineering Department of Case School, who outlined the purpose of the industrial conferences sponsored by the institution.

During the afternoon three technical papers were given. T. A. Weager, Cleveland District Manager for Buffalo Forge Co., presented a paper, entitled, The Use of Psychrometric Charts, in which he pointed out that there was hardly a product of manufacture which could not be advantageously produced in an air-conditioned atmosphere. He also explained the importance of air conditioning in the health and comfort of workers.

The subject of Automatic Regulation in Air Conditioning was presented by M. F. Rather, Cleveland Manager for Johnson Service Co., who described installations at Severance Hall and in a Cleveland department store, where air conditioning equipment is an important part of the mechanical equipment. The likelihood of using the Reversed Refrigeration Cycle for Summer and Winter Air Conditioning was the subject of a talk by R. E. Keyes, chief engineer of Cooling and Air Conditioning Corp. He described in detail the commercial application in a California office building.

On Thursday evening, with Dudley H. Scott, chief engineer of the Humphrey Co., Cleveland, acting as chairman, four papers were presented. Space Cooling with Ice was the subject of W. A. Schmid, research engineer for the City Ice and Fuel Co., Cincinnati, Ohio. The importance of cooling as a merchandising factor was discussed in the paper, Cooling for Profit, by M. K. Arenberg, Chicago manager of Ilg Electric Ventilating Co. He described the places where air cooling had proved advantageous in increasing the volume of business and predicted an increasing field of applications.

R. E. Hellmund, chief electrical engineer, Westinghouse Electric and Manufacturing Co., presented a paper, entitled, Problems of Power Supply for Air Conditioning, and S. B. Heath, development engineer for the Dow Chemical Co., discussed the subject, Dehumidification with Calcium Chloride.

The third session of the conference on Friday, March 18, was in charge of Ralph L. Harding, of Wilbur Watson and Associates, Cleveland, who introduced the first speaker, Dr. E. Vernon Hill, editor of *The Aerologist*, who described Standards of Air Quality, pointing out that the basis of all air conditioning for comfort was the data developed at the Research Laboratory of the American Society of Heating and Ventilating Engineers in co-operation with the U. S. Bureau of Mines and the U. S. Public Health Service. He also described some experiments sponsored by the Society at Harvard School of Public Health with relation to the comfort zone for summer and winter.

Natural Air Movements and Their Effect on Air Conditioning was the subject of a paper by G. L. Larson, head of the Department of Mechanical Engineering, University of Wisconsin, who explained the influence of the movement of large volumes of air and the effect of air leakage and workmanship in building construction on the operation of air conditioning systems.

Winter Air Conditioning and Human Comfort was discussed by F. C. Houghten, director of the Research Laboratory of the American Society of Heating and Ventilating Engineers, who summarized the results of many years investigations at the Society's laboratory in Pittsburgh at the U. S. Bureau of Mines, where studies have been made of the physiological reactions of human beings to various conditions of temperature, humidity and air movement.

Heat and Power Load Curves as Affected by Air Conditioning was the title of a paper given by J. H. Walker, superintendent of central heating, The Detroit Edison Co., who gave data on the operation of fully air-conditioned buildings using central heating in winter and cooling in the summer months.

At 2:30 on Friday afternoon, A. G. Trumbull, chief mechanical engineer of the C. & O., Pere Marquette, Erie and Nickel Plate Railroads, opened the meeting and introduced W. H. Driscoll, vice-president of Thompson-Starrett Co., who presented a paper on the Economics of Air Conditioning in Large Buildings.

The Air Conditioning of Railway Trains, by W. H. Carrier and R. W. Waterfill, was presented by Mr. Carrier, chairman of the board of the Carrier Corp., who described the several applications of air conditioning systems to railway cars and told of the experimental work carried on to meet the severe requirements of railway installation and operation.

Year 'Round Air Conditioning with the Individual Unit was discussed by W. D. Jordan, president of Air Control Systems.

On Friday evening an open meeting for members of Cleveland engineering societies was held at John Hay High School, with O. C. Sabin, president of the Cleveland Engineering Society, acting as chairman.

The first speaker was Dr. W. E. Wickenden, president of the Case School of Applied Science, who gave a splendid address, entitled, Cooperation Between the Engineering College and Its Graduates.

Atmospheric Pollution, Its Measurement and Control was given by H. B. Meller, chief of Bureau of Smoke Regulation, Pittsburgh, who said that for every million tons of bituminous coal burned in the City of Cleveland, 30,000 tons of soot, ash and tarry particles were thrown into the atmosphere. He pointed out that control of air pollution from smoke stacks must include not only elimination of visible smoke, but separation of other elements before discharge to the atmosphere.

The Importance of Air Conditioning in the Homes of America was explained by Prof. F. B. Rowley, president of the American Society of Heating and Ventilating Engineers and director of the Experimental Engineering Laboratories at the University of Minnesota, in his paper Air Conditioning for the Home.

Charting Human Comfort was given in a series of pictures, showing how the zone of comfort was determined by a long series of experiments at the Research Laboratory of the American Society of Heating and Ventilating Engineers in co-operation with the U. S. Bureau of Mines and the U. S. Public Health Service. This presentation was given by F. C. Houghten, director of the Research Laboratory of the A. S. H. V. E.

The concluding session of the Conference on Air Conditioning opened at 10:00 o'clock, Saturday morning, with F. B. Rowley, president of the American Society of Heating and Ventilating Engineers, as chairman.

H. C. Murphy, vice-president of American Air Filter Co. and chairman of the Technical Advisory Committee on Air Cleaning of the A. S. H. V. E., presented a paper, entitled, Atmospheric Dust and Air Cleaning Devices. A series of slides showed various types of dust and devices for measuring and eliminating dust in ventilating and air conditioning work. The speaker also listed the cities which have made improvement in atmospheric conditions by elimination of smoke and dust.

The Requirements for the Successful Unit Conditioner were described by R. E. Robillard, of Frigidaire Corp. This paper told of the developments in designing conditioning units for home and room installations.

F. Paul Anderson, dean of Engineering of the University of Kentucky, discussed The Human Power Plant. Dean Anderson compared the human body to the power plant and told how the various functions were comparable.

STANDING (LEFT TO RIGHT)
J. H. Walker, *Member*, A. S. H. V. E. Committee on Research; G. L. Larson, *Chairman*, A. S. H. V. E. Committee on Research; F. C. Houghten, *Director*, A. S. H. V. E. Research Laboratory; A. V. Hutchinson, *Secretary*, A. S. H. V. E.; Ralph L. Harding; G. L. Tuve, *Case School of Applied Science*.

SEATED (LEFT TO RIGHT)
F. H. Vose, Professor of Mechanical Engineering, Case School; E. D. Milener, *American Gas Association;* Dr. W. E. Wickenden, President, *Case School of Applied Science;* W. H. Carrier, *Past- President*, A. S. H. V. E.; F. B. Rowley, *President*, A. S. H. V. E.; E. Vernon Hill, *Past-President*, A. S. H. V. E.

Problems of the School and Public Building was the title of a paper contributed by Samuel R. Lewis, consulting engineer of Chicago, and presented by his son Edward Lewis.

The Gas Industry's Position in Relation to Air Conditioning was given by E. D. Milener, Industrial Research Representative of the *American Gas Association*.

In connection with these presentations, a large radio audience had the opportunity of hearing speakers over Station WTAM in Cleveland. The broadcasts were given on Thursday, Friday and Saturday evenings, respectively, by H. B. Meller, Pittsburgh, Prof. F. B. Rowley, president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and Dean F. Paul Anderson, Past president of the A. S. H. V. E. and former director of its Research Laboratory.

It was notable that among the papers presented 16 members of the A. S. H. V. E. participated in the program and included Pres. F. B. Rowley, Prof. G. L. Larson, chairman of the Committee on Research, F. C. Houghten, director of the Research Laboratory, five past presidents, and several other members, who have been active in the Society.

The Engineering Buildings and Laboratories of the Case School of Applied Science were open for inspection and a number of manufacturers of air conditioning equipment had their products on display. A. O. Willey was in charge of exhibits and arrangement of the sessions was under the direction of Prof. G. L. Tuve, of Case School of Applied Science, and Dudley H. Scott, of the Cleveland Engineering Society.

April Itinerary of President Rowley

April will be an active month for Officers of the Society according to present plans with a meeting of the Committee on Research scheduled for April 8 in Pittsburgh, followed by a meeting of the Council in the same city April 9.

An official visit to the Pittsburgh Chapter will be made by Pres. F. B. Rowley on Monday, April 11, and on Tuesday, April 12, he will be with the members of the Western New York Chapter in Buffalo. The Toronto members will entertain President Rowley at Toronto Wednesday, April 13, and on the 14th and 15th he will visit the Michigan Chapters in Detroit and Grand Rapids. The Wisconsin members will be his hosts on Monday, April 18.

During the month of March President Rowley and Secretary A. V. Hutchinson made visits to the Illinois Chapter in Chicago March 14, Kansas City Chapter on March 15, and the St. Louis Chapter March 16.

The Cincinnati members of the Society held a luncheon March 17, at the Cincinnati Club of Cincinnati, Ohio, for President Rowley and on Friday evening, March 18, the Cleveland Chapter held a dinner at Fenway Hall, Cleveland, in honor of his visit to the Conference on Air Conditioning at the Case School of Applied Science.

Publish Standards for Pipe Flanges and Fittings

The *American Standards Association* announces the publication by the *American Society of Mechanical Engineers* of the American Standard for Cast-Iron Pipe Flanges and Flanged Fittings for maximum non-shock working hydraulic pressure of 800 lb per sq in. (gage) at ordinary air temperatures. The standard was prepared by the committee on Pipe Flanges and Fittings under the sponsorship of the *American Society of Mechanical Engineers*, the *Heating and Piping Contractors National Association*, and the *Manufacturers Standardization Society of the Valve and Fittings Industry*.

Oil Burner Meeting and Show

The program for the meeting of the *American Oil Burner Association* to be held in Boston, April 11-16, has scheduled a great many subjects for discussion which will be of interest to men engaged in oil heating work, whether they follow engineering or merchandising. An attractive show will be held at Mechanics Hall and visitors will have an opportunity of learning about the results of cooperative research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Oil Burner Association* at Yale University. In Booth 32 the A. S. H. V. E. will graphically show some of its research work in the laboratory at the U. S. Bureau of Mines, Pittsburgh, and in several cooperating institutions.

The exhibits will open at 7:30 p. m., Monday, April 11, and daily thereafter from 1:30 to 10:00 p. m.

N. D. H. A. Proceedings

The 1931 Proceedings of the *National District Heating Association* has been issued to members. It covers the transactions of the 22nd Annual Meeting in June, 1931, and includes the complete technical papers and discussions, reports of officers and committees as well as a comprehensive index.

Local Chapter Reports

Kansas City

February 8, 1932. Twenty members and guests of the Kansas City Chapter held a meeting at the Ambassador Hotel, which was called to order by Pres. J. M. Arthur.

The unemployment situation among heating and ventilating engineers was discussed and Vice-President W. A. Russell was appointed to head a committee to work in conjunction with other Engineering Societies in the determination of suitable relief.

A committee was appointed to arrange to have the Chapter take a table at the dinner dance and represent Kansas City Engineers at the Washington Day celebration.

A letter from Herman Henrici was read, in which he expressed regret at having been unable to attend chapter meetings for some time on account of his health.

Considerable time was devoted to a discussion of membership both in the Chapter and national Society and to chapter relations. E. K. Campbell, Carl Clegg, N. W. Downes and Mr. Russell were appointed to thoroughly analyze the matter of membership and dues and make a suitable report.

The civic feature of the evening was a presentation by Mr. Charno, a local attorney, of the Value of the Boy Scout Organization to Society and to the Country. Mr. Charno brought a clear, forceful message, showing the necessity of a character building institution of this kind to offset the lawlessness and disintegration of today.

Mr. Campbell made a report on the Annual Meeting of the Society in Cleveland, discussing in particular the Report of the Committee on Ventilation Standards.

L. E. Evans, of the General Electric Co., was the principal speaker of the evening. He gave a most interesting paper covering phases of the application of electricity to the heating and ventilating industries, including generation, distribution, and utilization of electric power. A discussion followed on the advantages and limitations of different types of equipment and methods of application. The members of the Chapter felt that they were very fortunate in having Mr. Evans present to give them such a fine talk and lead the discussion so ably.

Western Michigan

February 8, 1932. One of the most interesting meetings since the organization of the Chapter was held in the English Room of the Rowe Hotel, Grand Rapids, with 32 members and guests present.

After dinner, the regular order of the meeting was changed to permit Corp. Ray Sullivan, of the Michigan State Police Dept., East Lansing, to give a short but interesting talk on Crime and Methods of Combating It. The Chapter hopes to have similar talks on non-technical subjects at future meetings.

Pres. O. D. Marshall then called the business meeting to order and, after reports by several committees, he gave a short resumé of his visit to Cleveland for the Annual Meeting of the Society.

The meeting was then opened for presentation of papers and discussion. Prof. L. G. Miller, of the Engineering College, Michigan State College, gave a summary of the paper presented by C. P. Yaglou at the Annual Meeting in Cleveland, entitled, Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods. Professor Miller in his talk gave evidence that he had taken a great deal of interest in Mr. Yaglou's paper and presented it to the meeting in a very interesting form. As a result, considerable discussion followed.

President Marshall introduced Prof. H. B. Dirks, Dean of Engineering College, Michigan State College, who summarized the paper, Application of Refrigeration to Heating and Cooling of Homes, which had been presented at the Annual Meeting by A. R. Stevenson, Jr., at a joint session with the *A. S. R. E.* Prof. Dirks was well informed on this paper and added to the interest by the use of blackboard illustrations. Although the subject was a difficult one to present, the speaker made excellent use of his teaching experience and an interesting discussion period followed.

The chapter gave a rising vote of thanks to Professor Miller and Professor Dirks and hopes that it may be fortunate enough to have them again as speakers.

The meeting was adjourned at 10:00 p. m.

A great many of the members and guests present were from nearby cities such as Kalamazoo, Lansing, Muskegon and Battle Creek, indicating that the meetings are creating interest.

Philadelphia

February 11, 1932. The regular meeting of the Philadelphia Chapter was held in the Auditorium of the Engineers Club. Dinner was served promptly at 6:30 p. m. and at its conclusion Pres. L. C. Davidson called the meeting to order.

The secretary read a resolution of condolence and regret, which had been prepared by a committee appointed by Past-President E. N. Sanbern and occasioned by the death of a devoted member, John J. Wilson.

The treasurer's report was submitted and approved.

President Davidson called upon the secretary to read the names of the members of the various committees appointed to serve during his term of office.

The meeting was then turned over to J. H. Hucker, chairman of the Meetings Committee, who introduced S. K. Wolf, manager of the Electrical Research Products, Inc., New York. Mr. Wolf gave a very interesting talk on the subject of, Noise: Its Measurement and Abatement, and showed slides illustrating the equipment that has been developed and is used for the measurement, location and recording of different sounds. The subject was one of interest to all members.

A rising vote of thanks was extended to Mr. Wolf and his talk was proclaimed one of the most interesting that has been presented to the Chapter for some time.

After a general discussion, the meeting was adjourned.

Michigan

February 15, 1932. S. S. Sanford, of the Detroit Edison Co. was the principal speaker at the meeting of the Michigan Chapter held in the Detroit Engineering Society Building. About 50 members and guests were present.

Mr. Sanford was introduced by Pres. G. H. Giguere and presented the paper, Field Studies of Office Building Cooling. Illustrating his talk with slides, Mr. Sanford discussed the air conditioning of the first sixteen floors and two basements of the Union Guardian Building. He also explained several methods of cooling small spaces, either by unit coolers, several of which are now on the market, or by means of ice. According to the speaker, a number of air conditioning installations will be made in Detroit during the coming year, in some of which the small cabinet units will be used and in one installation ice will be the medium employed. Although the use of small cabinet coolers is expensive at the present time, the cost will undoubtedly be reduced as their use increases. One instance was reported where a restaurant had increased its business 25 per cent by the installation of a cooling system.

E. P. Wells of the Union Guardian Building gave a brief description of the building operation.

The discussion of the paper showed a great interest in the subject and was participated in by Messrs. W. C. Randall, W. A. Rowe, J. F. McIntire, A. C. Wallich, J. R. McColl, H. E. Paetz, and others.

Pittsburgh

February 15, 1932. Pres. R. B. Stanger called the meeting of the Pittsburgh Chapter to order in the Grill Room of the Hotel Mayfair, with an attendance of 56 members and guests.

The minutes of the January meeting were read and approved.

President Stanger reported for F. C. McIntosh, chairman of the Program and Publicity Committee, who was prevented by illness from attending the meeting. He announced that W. H. Driscoll, a Past President of the Society, would address the chapter at the March meeting and Prof. F. B. Rowley, President of the A. S. H. V. E., would be the speaker at the April meeting.

The secretary read a report prepared by the treasurer, which showed a bank balance of $220.93.

E. C. Evans, chairman of the Membership Committee, stressed the value of the A. S. H. V. E. to men eligible for membership and appointed the following men to serve on his committee: P. A. Edwards, S. H. Harper, F. H. Hecht, G. S. McEllroy, H. L. Moore, L. B. Pittock, R. J. J. Tennant, and E. S. Tower.

President Stanger presented Press C. Dowler, president of the Pittsburgh Chapter of the *American Institute of Architects,* who, in turn, introduced the speaker of the evening, Dr. Hubert C. Eicher, Director of the School Building Division of the Pennsylvania Department of Public Instruction.

Dr. Eicher spoke on Some Local and State Problems in Heating and Ventilating School Buildings. He said he considered misunderstanding to be chiefly responsible for the friction and strife which exist among engineers, architects, school and public officials. Dr. Eicher traced the history of Pennsylvania school building codes, showing how they are applied and how they have been changed. In 1911 a law placed all plans and specifications of school buildings within the jurisdiction of the State Board of Public Education. In 1921 this code was revised with the assistance of the Pittsburgh and Philadelphia Chapters of the A. S. H. V. E. With a few slight changes in 1924 this revised code has served Pennsylvania admirably until the present time. This year a revision of the code is being prepared by nine experts in different branches and it is expected that a sane and rational solution will be found to the problems with which the engineer, architect, school and public officials, and school children have to cope.

The members and guests extended a rising vote of thanks to Dr. Eicher in appreciation of his excellent talk.

Prior to adjournment, it was voted that the Board of Governors be given authority to set the date of the March meeting to conform with the plans of the speaker, previously mentioned.

Cleveland

February 23, 1932. Thirty-seven members and guests assembled in the rooms of the *Cleveland Engineering Society* for a meeting of the Chapter, at which Pres. W. E. Stark presided.

The following members were unanimously elected to membership in the Cleveland Chapter: C. A. Wheeler, J. N. Schweikert, and J. J. Hendrickson.

T. A. Weager, general chairman of the Committee on Arrangements for the Annual Meeting of the Society, which was held in Cleveland in January, reported for the various committees and read the report of the chairman of the Finance Committee. After some discussion it was voted that this report be accepted. Mr. Weager also spoke of letters received from and comments made by various members who attended the meeting.

A rising vote of thanks was given to Mr. Weager and his associates for carrying the meeting through to a successful conclusion.

Prof. G. L. Tuve, of Case School of Applied Science, was introduced by President Stark and spoke of the Conference on Air Conditioning to be held March 17-19 and sponsored jointly by the *Cleveland Engineering Society* and Case School of Applied Science.

President Stark suggested that the March meeting of the Chapter be in the nature of a dinner meeting to be held as arranged, with a number of visiting speakers as guests. There was some discussion regarding a method of helping finance the program.

Colonel Schweikert, chairman of the Unemployment Committee, spoke of the activities of the various bodies working under his direction and requested an appropriation of $10.00, as the contribution of the Cleveland Chapter toward the amount required for legalization of the Employment Bureau. This request was granted.

The ideals of the Society were expressed and a motion was made by Mr. Weager and carried that collectively and individually the Unemployment Committee would be supported by the members of the Society by requesting that members do no outside work of an engineering nature, if they were regularly employed.

The speaker of the evening was R. M. Conner, Director of the Laboratory of the *American Gas Association,* whose subject was, What the Gas Industry Has to Offer the Heating (and Cooling) Engineer. Mr. Conner outlined the history of the gas industry, gave statistics regarding its rapid growth, and traced the history of house heating since its early beginning. He discussed the history, methods, and progress in the use of air conditioning for house heating and gave interesting information on the use of silica gel, discussing the research being done by the *A. G. A.* along these lines. Mr. Conner showed illustrated slides, depicting the work of the *A. G. A.* Laboratories, including their method of testing, and spoke in detail of some of the tests that are made.

After some discussion the meeting was brought to a close with a vote of thanks for the speaker.

Southern California

February 15, 1932. A dinner and business meeting of the Southern California Chapter took place at the Engineers' Club, Los Angeles.

O. W. Ott, chairman of the Technical Ordinance Committee, reported the results of a meeting held by his committee with Mr. Baccus of the City regarding an ordinance pertaining to changes in the building and safety code.

J. S. Louis presented a paper on Double Suction and Multiple Effect Compression in Refrigeration. Mr. Louis' talk was illustrated by slides and showed various designs and methods of refrigerating.

J. Frank Park and Sydney Locke spoke on Centrifugal Refrigeration. Mr. Park's talk was illustrated by slides and a motion picture showing installations and traced the development of mechanical refrigeration from its inception. Mr. Locke gave a short discussion on Refrigeration by Steam.

At the March meeting of the Chapter, Ralph E. Phillips and Donald T. Robbins, as co-authors, will present a paper on Air Conditioning and Its Relation to Public Buildings. As Mr. Phillips and Mr. Robbins are the authors of a number of articles which have been published recently, a very interesting meeting is anticipated.

Massachusetts

February 11, 1932. The February meeting of the Massachusetts Chapter took place at 12:30 p. m. at Durgin Park's on Hayward Place, Boston, where luncheon was enjoyed by 40 members and guests.

H. E. Morton, of Boston, spoke to the gathering on Refrigeration Application to Air Conditioning and the Use of the Refrigeration System for Heating. Mr. Morton illustrated his talk by lantern slides and gave a very comprehensive idea of the work being done in this field.

Metropolitan Life Issues Pamphlet on Air Conditions and the Comfort of Workers

A very interesting pamphlet entitled Air Conditions and the Comfort of Workers has recently been issued by the Policyholders Service Bureau of the Metropolitan Life Insurance Company, New York. This is Industrial Health Series No. 5, and the contents include Temperature Standards for Comfort, Measurement of Physical Properties of Air, Conditioned Air, Effective Temperature, The Comfort Zone, The Effects of High Atmospheric Temperatures on Man, Removal of Excessive Heat and Moisture, Air Pressure and Air Cleanliness.

The pamphlet quotes a considerable amount of the comfort data and other similar research results obtained by the Society at the Pittsburgh Laboratory, and also at the Harvard School of Public Health, under the cooperative agreement with that institution. The subject matter is treated in a very clear and concise manner, reduced to such a form that it would be readily comprehensible to the layman.

This is one of many publications issued by the Policyholders Service Bureau of the Metropolitan Life Insurance Company in the interest of better management in business.

New Officers of Frost Research Laboratory

An expansion of the activities of the Frost Research Laboratory is indicated in the election of new officers at its annual meeting which was held in Norristown, Pa., February 24, 1932. R. V. Frost, a member of the Society, who established the laboratory in 1925 and who has directed its activities since that time, continues as president and director. The other officers are A. S. Armagnac, vice-president, also a member of the Society, and Gustave Petersen, secretary-treasurer. Mr. Armagnac and Mr. Petersen were for many years associated as publishers of the *Heating and Ventilating* magazine. They will take an active part in the business management and promotion of the laboratory.

The laboratory will continue to specialize in the testing of heating boilers, domestic stokers, radiators, oil and gas burners, convectors and air conditioning devices. For the latter service, the laboratory maintains a fully-equipped calorimeter test room in which any ordinary condition of temperature and humidity can be maintained, including the simulation of a rain storm.

Some of the largest industrial concerns in the country have been served by the laboratory which has been successful in supplementing established research organizations in the development of new heating products and in the preparation of accurate test data for catalog purposes.

A New York office of the laboratory has been opened at 30 Church Street, where Mr. Armagnac and Mr. Petersen will maintain their headquarters.

W. B. Crawford Dies

Members of the Society and friends of William Blake Crawford will be saddened to learn of his death on February 20, 1932, after a short illness.

Mr. Crawford was born at Oshkosh, Wis., on December 25, 1876, but spent the greater part of his life in Chicago. From 1896 until 1924 he held the positions of apprentice, draftsman, manager, sales and designing engineer with Crawford Co., Garden City Fan Co., Illinois Blower & Sheet Metal Works, Consolidated Engineering Co., Arnold Co., and Massachusetts Blower Co. In 1924 he became associated with Jas. P. Marsh & Co. as consulting engineer and later as manager of their specialty and trap departments.

Mr. Crawford served also as sales manager for Terre Haute Boiler Works Co. and at the time of his death was combustion engineer for the Seneca Petroleum Co.

In 1921 Mr. Crawford became a member of the American Society of Heating and Ventilating Engineers and the Officers and Council extend their sincere sympathy to his widow who survives.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 16 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by April 15, 1932, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Balsam, Charles P., Mgr., Refrig. & House Heating Div., Washington Gas Light Co., Washington, D. C.	E. J. Febrey S. F. Gardner	T. H. Urdahl G. C. Breitenbach
Bowers, Ross C., Sales Mgr., Minneapolis-Honeywell Regulator Co., Milwaukee, Wis.	E. A. Jones H. F. Haupt	A. M. Wagner C. H. Randolph
Cohen, Philip, Acting Dist. Mgr., B. F. Sturtevant Co., Cleveland, Ohio.	W. E. Stark R. G. Davis	W. C. Kammerer T. A. Weager

DEAN, CHARLES LYMAN, Asst. Prof., Mech. Engrg., University of Wisconsin, Madison, Wis.	G. L. Larson D. W. Nelson	F. G. Weimer W. H. Wilson
GILMOUR, ALAN BOWDOIN, Salesman, B. F. Gilmour Co., Inc., Brooklyn, N. Y.	E. W. Mandeville M. W. Ehrlich	F. W. Phillips, Jr. E. G. Hinchman
GREENBURG, LEONARD, Assoc. Dir. of Research, Pierce Foundation, New Haven, Conn.	W. H. Carrier Thornton Lewis	D. D. Kimball V. J. Cucci
HALL, JOHN R., U. S. Blower & Heater Co., Minneapolis, Minn.	A. B. Algren F. B. Rowley	H. E. Gerrish M. S. Wunderlich
KONZO, SEICHI, Special Research Associate, University of Illinois, Urbana, Ill.	A. C. Willard A. P. Kratz	E. O. Eastwood E. B. Langenberg
MCGONAGLE, ARTHUR, Consulting Engineer, Arthur McGonagle, Pittsburgh, Pa.	J. I. Lyle J. F. Hale	L. C. Soule R. B. Stanger
MOREAU, DONATO, Manager, Heating & Ventilating, J. Knox Corbett Lumber Co., Tucson, Ariz.	H. A. Nelson O. W. Ott	C. M. Barker C. E. Johnson
POLLARD, ALFRED L., Genl. Supt., Steam Heat Dept., Puget Sound Power & Light Co., Seattle, Wash.	W. W. Cox C. H. Eckart	William Mallis S. C. Heath
PRYIBIL, PAUL LESTER, Hucker-Pryibil Co., Philadelphia, Pa.	J. H. Hucker W. P. Culbert	E. N. Sanbern M. F. Blankin
STEVENSON, ALEXANDER R., JR., Staff Asst. to Vice-Pres., General Electric Co., Schenectady, N. Y.	W. H. Carrier Dr. C. W. Brabbée	Elliott Harrington F. H. Faust
TENNEY, DWIGHT, Chief Engr., Booth Engrg. Co., Ltd., Newark, N. J.	A. E. Stacey, Jr. Harry Bentz	
WEBER, C. A. M., Manager, Motored Appliance Engrg. Dept., Westinghouse Electric & Mfg. Co., Springfield, Mass.	Frank Thornton, Jr. M. M. Timmerman	S. M. Kintner—*AIEE* R. E. Hellmund—*AIEE*
WINSLOW, CHARLES-EDWARD A., Prof. of Public Health, Yale School of Medicine, New Haven, Conn.	W. H. Carrier Thornton Lewis	D. D. Kimball V. J. Cucci

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

BRILL, JOSEPH W., M. G. Lippincott, Hornell, N. Y. (*Advancement*)

BULLEIT, CHARLES R., Manager, Heating Dept., Plumbers Supply Co., Evansville, Ind. (*Advancement*)

CARLSON, EVERETT E., Branch Mgr., The Powers Regulator Co., St. Louis, Mo. (*Advancement*)

EASTMAN, CARL B., Branch Mgr., C. A. Dunham Co., Philadelphia, Pa. (*Advancement*)

FENTON, FREDERICK A., Asst. Branch Mgr., Richardson & Boynton Co., New York, N. Y.

HILLIARD, CHARLES ERNEST, Heating & Ventilating Engr., E. C. Hilliard, South Boston, Mass. (*Advancement*)

JACKSON, ALTON BLANCHARD, New England Dist. Mgr., Carrier-York Corp., Winchester, Mass.

KEELER, ANSON H., Sales Engr., Vento & Arcoblast Dept., American Radiator Co., Philadelphia, Pa.

KERSHAW, MELVILLE G., Ventilating Engr., E. I. duPont de Nemours & Co., Wilmington, Del. (*Advancement*)

MACMAHON, WILLIAM KENNEDY, Asst. Mgr., Utilization Dept., Washington Gas Light Co., Washington, D. C.

MILLER, MERL WILLIAM, Manager, Unit Heater Dept., The Trane Co., La Crosse Wis. (*Advancement*)

PHELPS, HAROLD ROY, Mech. Engr., American Blower Corp., Detroit, Mich. (*Advancement*)

SCHLICHTING, WALTER G., Mgr., Air Conditioning Dept., Clarage Fan Co., Kalamazoo, Mich.

TUCKERMAN, GEORGE E., Mgr., Air Conditioning Div., York Ice Machinery Corp., Philadelphia, Pa.

WEBSTER, WARREN, JR., Vice-Pres., Treas., Warren Webster & Co., Camden, N. J. (*Advancement*)

ASSOCIATES

CLODFELTER JOHN LUTHER, Supt., Carolina Sheet Metal Corp., Philadelphia, Pa.

FILKINS, HARRY L., Vice-Pres., City Ice Co. of Kansas City, Kansas City, Mo.

JOHNSTON, JOHN F., JR., Johnston Bros., Inc., Ferrysburg, Mich.

REID, HERBERT F., Partner, Reid-Graff Co., Muskegon Heights, Mich.

WALLON, CHARLES D., Sales Mgr., Spencer Foundry Co., Ltd., Toronto, Ont., Canada.

WALSH, CLIFFORD J., Johnston Bros., Inc., Ferrysburg, Mich.

JUNIORS

BLACK, WILLIAM B., Salesman and Engineer, Bryant Gas Heating, Inc., Pittsburgh, Pa.

PAUL, DONALD I., Gurney Foundry Co., Toronto, Ont., Canada.

ROCKWELL, THEODORE F., Instr., Carnegie Inst. Tech., Pittsburgh, Pa.

STUDENTS

FEYDT, JOHN CHARLES, Student, Carnegie Inst. Tech., Pittsburgh, Pa.

FITZSIMONS, J. PATRICK, Student, Carnegie Inst. Tech., Pittsburgh, Pa.

HICKEY, JAMES WILLIAM, Student, Carnegie Inst. Tech., Pittsburgh, Pa.

VINSON, NEAL LORING, Student, Carnegie Inst. Tech., Pittsburgh, Pa.

Heating · Piping and Air Conditioning

EDITORIALS

» » In an address at a meeting of the New York sections of the *A. S. T. M.*, *A. I. M. M. E.*, and the *A. W. S.*, H. F. Moore, research professor of engineering at the University of Illinois, pointed out that in recent years two new limits of strength of materials have been reported from many testing laboratories. The first of these is the *endurance limit* or *fatigue limit* and the second the *creep limit*.

"The significance of the fatigue limit is a matter of considerable debate," said Professor Moore. "It would seem at first that this limit has a rather direct bearing on the ability of a material to withstand repeated stress. Owing to the fact that for steel this limit seems to be fairly closely correlated with the tensile strength, the fatigue test has not as yet made its appearance in specifications to any great extent, but resistance to repeated stress is judged from the tensile strength. It should *not* be judged by any kind of elastic limit.

"It is in the field of machinery that most fatigue failures occur, and even in this field, it should be emphasized that many machine parts are in no particular danger of failure by fatigue. Many pressure vessels have the load on them repeated only a few hundred times, which hardly brings fatigue strength into play. The case of the steam boiler where the stress may be repeated a few thousand times in the life of the boiler seems a border-line case. We may say with reference to the fatigue limit as with reference to any other proposed new test, 'Do not use the test unless it seems to give results closely correlated with the service required.'

"The second new limit is the *creep limit*. At elevated temperatures for ordinary metals under a steady load there goes on a continuing stretch or compression of the metal. This is different from the plastic flow which takes place at the yield point, for such plastic flow does *not* continue indefinitely under steady load. Whether for any metal there exists a limit below which the creep is zero for a given temperature is as yet uncertain. However, a limiting stress for most metals and most temperatures can be found below which the amount of creep will not be disastrous in the desired life of the member. This whole subject of creep is such a recent development that we can not say how important this test will be or whether any short-time tests for this creep will become feasible. It gives, however, promise of very great significance as a test value in connection with metals which have to carry stresses at high temperatures."

» » Heating systems are installed for the purpose of maintaining the desired temperature in rooms, enclosures and buildings. Modern equipment for supplying and controlling the amount of heat needed is the result of the progress that has been made by the heating engineer.

One of the principal requirements of a heating system, is that the heat should be supplied and controlled *economically*. The economical operation of a heating system depends to a large extent on the construction of the building. A well-designed heating system may be expensive to operate if proper provisions for conserving heat are not made by the architect.

Good workmanship and properly insulated buildings, together with other means for preventing heat losses, should be the aim of all parties connected with the development of a building. As in most engineering problems, cooperation will result in profit to all concerned.

» » A few days ago an air conditioning engineer was asked by a sales-manager to talk to his salesmen, who had gathered for a conference. The engineer found that what was wanted was a set of easily applied rules for designing air conditioning systems so that any member of the sales force, regardless of his technical training or experience, could design and sell such plants without referring the matter to an engineer.

It was pointed out that in air conditioning, each job is an engineering problem in itself. The sales-manager countered with the statement that "So-and-so's salesmen lay out their own air conditioning jobs, so why can't we do it?"

It is true that So-and-so's salesmen (who are not engineers) are making such lay-outs but it is known also that systems put together without engineering knowledge are usually quite unsatisfactory. Some are acceptable only because the buyers themselves don't know what to expect of a true air conditioning system.

Unless such practices are checked a reaction that will be unfavorable to air conditioning seems likely.

"Open for Discussion"

Determining Total, Static and Velocity Pressure

By L. A. Harding*

THE true meaning of the indicated or observed readings of the pressure or head measuring device or gage when connected with either a static or impact tube located in the pump or fan suction is so well understood by engineers, that it would appear almost superfluous to enter into a discussion of the matter. The writer is not aware of any cases of misunderstanding which may have arisen in practice in regard to the true meaning or application of observed suction-side gage readings.

The gage as customarily employed actually indicates a difference in pressure between the atmospheric pressure and the pressure within an enclosure (usually a pipe or duct) at the point of measurement either above or below atmospheric pressure at the time and place of measurement.

The writer agrees with A. A. Berestneff[1] that certain definitions appearing in the text book to which he refers could be improved. The definitions given by Mr. Berestneff, however, do not convey a very adequate meaning of the terms defined. The following definitions for velocity, static and total head or equivalent pressure may or may not be considered as an improvement in this connection.

Velocity Head or Equivalent Pressure is interpreted to mean the head or equivalent pressure required to accelerate the media under observation, from a state of rest to the velocity attained at some particular point in the pipe, duct or enclosure being considered.

The velocity head or equivalent pressure at any particular point is *ordinarily* determined by taking the difference between the gage reading connected with or attached to a pitot tube and a piezometer, both located at the point considered. The numerical value of the velocity head or equivalent pressure is obviously accompanied by a plus (+) sign.

The *average* velocity head over the duct cross-section at the point considered is what is customarily inferred, when the term *velocity head* is employed in engineering calculations, pipe or duct friction tables, etc.

Static Head or Equivalent Pressure is quite generally interpreted to mean the observed reading of a gage connected with or attached to a piezometer. This gage actually indicates the difference in the head or equivalent pressure between the atmospheric pressure and the absolute radial pressure within the pipe, duct or other enclosure in which the piezometer is located.

It is evident that the observed head or equivalent pressure so measured may be either above or below the atmospheric pressure. It is further evident that the observed static pressure may cause the material of the duct or pipe to be placed either under a compressive or a tensile stress. Static pressure does not, in the case of a suction pipe or duct, tend to "burst" the pipe in the usual interpretation of the word. The numerical value of the reading obtained in the manner described is accompanied by a plus (+) or minus (—) sign to indicate that the pressure is either above or below the atmospheric pressure, as the case may be.

The static head or equivalent pressure may obviously be recorded in absolute units in which case the numerical value of the pressure would always be accompanied by a plus (+) sign.

Total Head or Equivalent Pressure is quite generally interpreted to mean the *observed* reading of a gage connected with or attached to a pitot or impact tube. This gage actually indicates the difference between the absolute total head or equivalent pressure and the atmosphere head or equivalent pressure at the particular *point* of measurement within the pipe, duct or enclosure in which the pitot tube is located.

It is evident that the observed head or equivalent pressure difference may be either above or below the atmospheric pressure. The numerical value of the reading obtained in the manner described is accompanied by a plus (+) or minus (—) sign to indicate a pressure either above or below the atmospheric pressure, as the case may be.

The total head or equivalent pressure may obviously be recorded in absolute units in which case the numerical value of the pressure recorded would always be provided with a plus (+) sign.

The *correct* total average pressure at any point in a duct system can only be ascertained by calculation and not by a single reading of the pitot tube gage.

The *average* velocity at the point considered is first ascertained by the usual method of pipe or duct traverse with the pitot tube and the piezometer; the readings being taken on a circle drawn through the center of each concentric zone of equal area. The actual total pressure is then equal to the sum of the static pressure plus the pressure equivalent of the average velocity ascertained as above indicated.

Temperature or density corrections are obviously required when ascertaining pressure losses through any apparatus or system which changes the temper-

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[1] See "Total, Static and Velocity Pressure," by A. A. Berestneff, HEATING, PIPING AND AIR CONDITIONING, March, 1932, p. 195.

ature of the gas or air flow between the points considered.

The *average* total head at any point in a duct system on the delivery side of a fan is equal to the sum of all losses of head beyond the point considered in the direction of flow, plus the final average velocity head.

The *average* total head at any point in a duct system on the suction side of a fan is equal to the sum of all losses of head between the duct inlet and the point of measurement, and includes the loss of entry to the duct.

Obviously, if the delivery duct is connected to a room or chamber in which a static pressure is to be maintained above atmospheric pressure, this would necessarily have to be added.

Fan Connected to Inlet and Outlet Ducts

Fig. 1 shows a fan to which is connected a duct system to inlet and outlet and with the same assumed velocity pressures and the same pressure losses in each of the sections of duct as indicated by Mr. Berestneff in his article.

(1) If the suction duct is omitted, the pressure loss diagram is shown by Diagram *a*. The total pressure to be generated by the fan is evidently 2 in.
(2) If the delivery duct is omitted, the pressure loss diagram for the suction duct is shown by Diagram *b*. The total pressure to be supplied by the fan is evidently 1½ in.

In either case, the total fan pressure required is equal to "the sum of the friction losses plus the *final* velocity pressure."

If one desires, he may also state that the total pressure required to accelerate the air from a state of rest to its final velocity and overcome the duct resistance is equal to the difference between the total absolute pressure on the delivery and suction sides of the fan (measured at the fan).

For *a* the total pressure is equal to: 409"—407" = 2".
For *b* the total pressure required is: 407½"—406" = 1½".

One has a choice of a number of statements which will convey exactly the same meaning to anyone conversant with Bernoulli's energy equation of maintained flow in a duct or pipe system connected to a fan.

If the fan is connected to both a system of suction and delivery ducts, the total fan pressure required is equal to "the sum of the friction pressure losses plus the final velocity pressure," which in this case, referring to Diagrams *a* and *c* is:

(½" + ¼" + ¼" + ¼" + ¼" + ½") + 1" = 3".

or taking the difference between the absolute pressures on each side of fan, we have 409 — 406 = 3 in.

The pressure loss between any two *points* in any duct system (either suction or delivery ducts) is correctly given by the difference between the readings of the gage attached to the pitot tube, observing the correct + or — sign, or is the difference between the absolute heads or equivalent pressures at the points measured.

Referring again to Diagram *b*, the reading of the gage attached to the piezometer is 1½ in. The total fan pressure required is also 1½ in. This would only be true when the fan inlet and outlet velocities were equal, as is assumed in the preceding examples.

The total head to be supplied by the fan, however, is not (in the case of a suction duct) the same thing as the total head for the duct system, as will be observed from the pressure diagram *b*. The fan in this case must supply the additional head (½ in.) required to discharge the air corresponding to the velocity of the fan outlet. In this example, the area of the fan inlet and outlet were assumed, for convenience, to be equal. Some makes of induced draft fans *are* provided with equal inlet and outlet areas.

Mr. Berestneff is correct in his statement as to the obvious error in the text to which he refers in reference to the reversal of the true meaning of the observed readings of the pitot and piezometer gages on a suction duct. The pressure diagram to which the discussion is intended to apply is, however, correct.

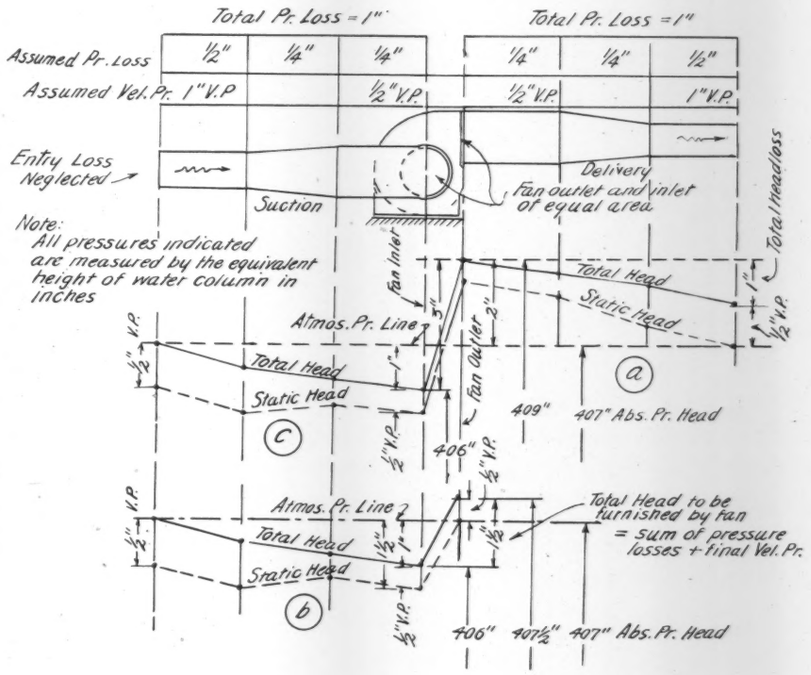


FIG. 1—FAN TO WHICH IS CONNECTED A DUCT SYSTEM TO INLET AND OUTLET

If suction duct is omitted pressure loss diagram is shown by Diagram *a*. If delivery duct is omitted, pressure loss diagram is shown by Diagram *b*.

Practical Piping Problems

Utilizing Heat of Condensate in Pipe Coils

▼

Draining a Pipe Coil

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Fig. 1 shows a method of installing pipe coils as heating surface that utilizes the heat from the condensation. The water of condensation accumulating in the trap is at a high pressure and temperature. Upon being discharged, pressure is relieved and part of the water will flash into steam at a lower pressure. This steam is used for heating in the second coil, located on the discharge side of the trap. The amount of heating surface in these pipes is governed by the size and length of the pipes containing steam under pressure in the first and second coils with the corresponding amount of condensation that may be discharged under the different conditions and room temperatures. It is better to provide an ample amount of surface for the trap discharge condensation, in order that as much heat will be radiated as possible, even if portions of these pipes are not fully heated. The size of the pipes heated by the trap discharge may be increased in order to contain more of the condensation. The discharge pipe from the trap is connected to the bottom pipe of the secondary coil in order that the piping will remain full of the heated water after the trap discharges, instead of the water flowing out of the pipes while heated. A tee with a plugged opening is placed at *C* for a clean-out. The location of the two different sections of these pipe coils may be in different places, as for example, one part may be placed on one side of the room and the other part on another, or wherever heat is desired.

In general this method of heating is wasteful practice. Consideration should be given to arranging for some method of returning the condensation to the boiler plant; a small vacuum pump can be used to good advantage. In some plants, however, the proper installation of return lines may be impossible; in such cases the use of the heat in the condensation should be the aim and the method mentioned may be utilized.

Fig. 2 shows a large pipe coil of the angle pattern used for heating a room for drying painted material. The pipe coil is made of 1½-in. pipe connected with branch tees or headers. The portion of the coil *A* was placed on the floor of the room and part *B* extended upward at one of the walls. The steam supply is at *C* and the return to the trap connection is at *D*.

Working conditions caused this heating coil to be put out of service during the winter season. To protect the piping from being damaged by freezing weather it was necessary to drain and blow it out. The supply and

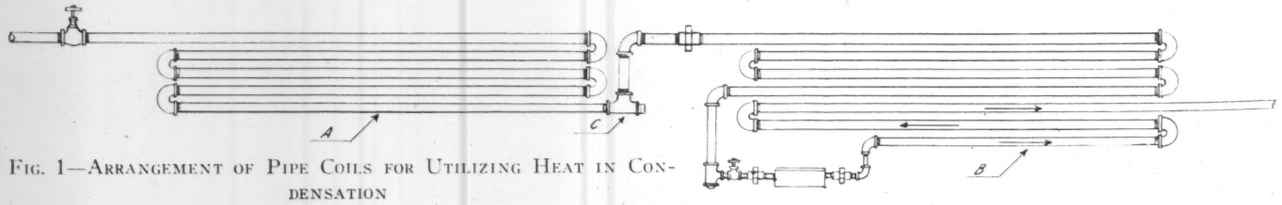


FIG. 1—ARRANGEMENT OF PIPE COILS FOR UTILIZING HEAT IN CONDENSATION

return were disconnected, a hose was attached at *C* and the piping blown out by using compressed air pressure. The maintenance man was not convinced that the water in the piping was completely removed. The plug at *E* was then removed, an additional air connection was made and the blowing out repeated, using both hose connections. A large amount of water was blown out in this way.

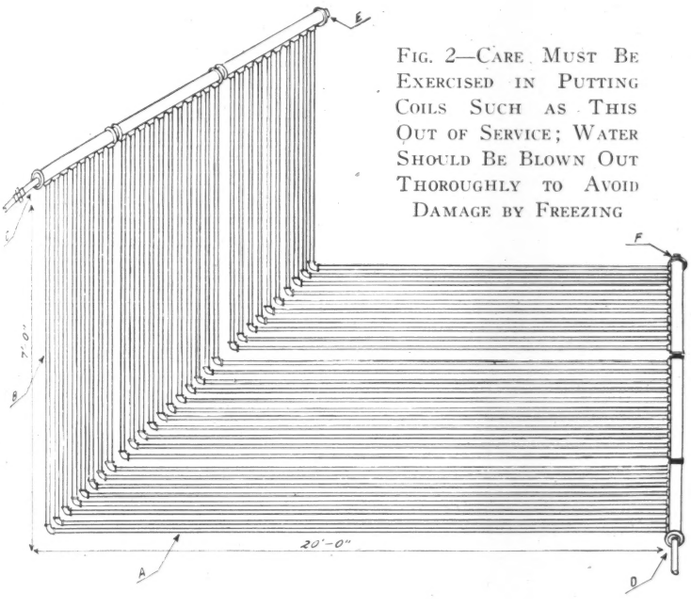


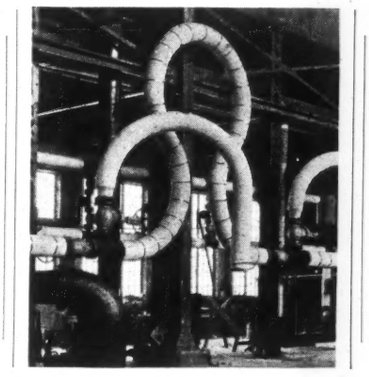
FIG. 2—CARE MUST BE EXERCISED IN PUTTING COILS SUCH AS THIS OUT OF SERVICE; WATER SHOULD BE BLOWN OUT THOROUGHLY TO AVOID DAMAGE BY FREEZING

This is mentioned to show that extreme care should be taken to remove all of the water that accumulates in piping, preparatory to placing equipment of this kind out of service, especially where floor coils are not provided with the proper amount of pitch to insure gravity drainage.

It is also recommended that a floor coil of this kind, consisting of a large number of pipes, should have an additional return connection at *F*, with a suitable steam trap at each return connection, *D* and *F*.

The condensation should be thoroughly removed from the floor coil in steam drying rooms and boxes. The heat passes upward, through and around the material to be dried, which is suspended in the dry room.

The section of the heating coil on the floor is the important part of the heating unit. The removal of the condensation in a proper manner will bring better results in the heating of the enclosure.



Preventing Drip from Pipes

By Herbert Chase

Pipes carrying cold water or other cold liquids through rooms which are humid or into which much steam escapes often cause a more or less regular dripping which is annoying and may result in damage to goods or equipment. The drip is the result, of course, of condensing water from the atmosphere. To prevent the condensation it is necessary to insulate the pipe in such a way that the heat, which is absorbed by the cold water when water vapor in the air comes in contact with the pipe, cannot reach the pipe.

A convenient method of accomplishing this is to paint the pipe with a heat-insulating material through which heat released by the vapor in the air cannot pass. The material in the paint ought to form a rather thick film and one which will not crack or peel readily, if lasting results are to be secured. In any case the film should be a good enough insulator to prevent the passage of heat at a rate which will permit drops of water to form and fall from the pipe.

Painting Black Pipe

Cork is one material that can be used for this purpose. The pipe to be painted must be clean, free from rust, and dry. If it is a black pipe, it should receive two coats of rust-inhibitive primer, such as basic lead chromate ground in a mixture of linseed and chinawood oil. The first coat should dry thoroughly before the second is applied and the second should also be well dried before the insulation is applied. Cork in dry granulated form can be mixed with the liquid vehicle or, if conditions permit, can be sprinkled on the pipe dry after the vehicle has been applied. The vehicle should be of a type that will not become very hard for some time; it may be made from a mixture of 30 per cent boiled linseed oil and 70 per cent of fish oil, to which is added a good turpentine dryer in the ratio of one pint of dryer to a gallon.

This finish is somewhat rough, which helps to prevent drops of water from forming and running together if the coating is not quite thick enough to prevent entirely the formation of moisture on the pipe. Should the color given by this coat be unsatisfactory, it can be followed by a size and then by a good paint or enamel.

Cleaning Galvanized Surfaces

For galvanized iron pipes a different procedure should be followed, as paint films of the ordinary type are more likely to peel from such a surface. The following procedure applies, so far as cleaning and priming is concerned, to any galvanized surface, whether on sheet metal, pipe, or other surfaces.

Cleaning of galvanized iron may be done with the following solution, which, because of its acid content, should be mixed and kept in glass or crock containers: Toluol, 70 per cent; denatured (ethyl) alcohol, 20 per cent; carbon tetrachloride, 5 per cent; and commercial muriatic (hydrochloric) acid, 5 per cent. This cleaner should be applied sparingly with a brush, but enough should be used to wet the entire surface without spattering or applying enough to drip on anything that will be attacked by the acid. It evaporates quickly and the

acid etches or roughens the surface enough to help the finish stick. The other ingredients are solvents which cut any oil or grease that may be on the surface and tend to loosen dirt. Work should not be done near an open flame and only in areas where ventilation is good.

When the cleaner has evaporated the surface should be washed thoroughly with clean water to remove any traces of acid that remain. When the surface is clean and dry it should be primed with linseed oil primer in which the pigment is 80 per cent zinc dust and 20 per cent zinc oxide. The linseed oil used in this mixture should have a low acid number, as it is termed, for best results. The primer may be purchased ready-mixed from a paint manufacturer.

Galvanized pipes treated in the manner indicated are ready for the cork-fish-oil-linseed-oil mixture referred to above, or may be painted with any good paint or enamel.

Conventions and Expositions

American Oil Burner Association: Annual convention, April 11-16; Mechanics Building, Boston, Mass. Secretary, H. F. Tapp, 342 Madison ave., New York City.

Bituminous Coal Conference: Fifth annual conference at Purdue University, April 14-15. Secretary, W. A. Knapp, Engineering Experiment Station, Purdue University, Lafayette, Ind.

American Welding Society: Annual meeting, April 27-29; New York City. Secretary, M. M. Kelly, 33 W. 39th st., New York City.

Heating and Piping Contractors National Association: Annual convention, May 16-18; Book Cadillac Hotel, Detroit, Mich. Secretary, Joseph C. Fitts, 50 Union Square, East, New York City.

National Warm Air Heating Association: Annual convention, May 18-19; Deshler-Wallick Hotel, Columbus, Ohio. Secretary, Allen W. Williams, 3440 A. I. U. Bldg., Columbus, Ohio.

American Boiler Manufacturers Association: Annual convention, May 23-25; Greenbrier Hotel, White Sulphur Springs, W. Va. Secretary, A. C. Baker, 801 Rockfeller Bldg., Cleveland, Ohio.

National Electric Light Association: Annual convention, June 6-10; Atlantic City Auditorium, Atlantic City, N. J. Secretary, A. J. Marshall, 420 Lexington ave., New York City.

National District Heating Association: Annual convention, June 14-17; William Penn Hotel, Pittsburgh, Pa. Secretary, D. L. Gaskill, Greenville, Ohio.

American Society for Testing Materials: Annual meeting, June 20-24, Atlantic City, N. J. Secretary, C. L. Warwick, 1315 Spruce st., Philadelphia, Pa.

American Society of Heating and Ventilating Engineers: Semi-annual meeting, June 27-29; Milwaukee, Wis. Secretary, A. V. Hutchinson, 51 Madison ave., New York City.

American Society of Mechanical Engineers: Spring meeting, June 27-July 1; Bigwin Inn, Lake-of-Bays, Ontario, Canada. Secretary, Calvin W. Rice, 33 W. 39th st., New York City.

INTERESTING Equipment . . . Developments

Railway Air Conditioning Units

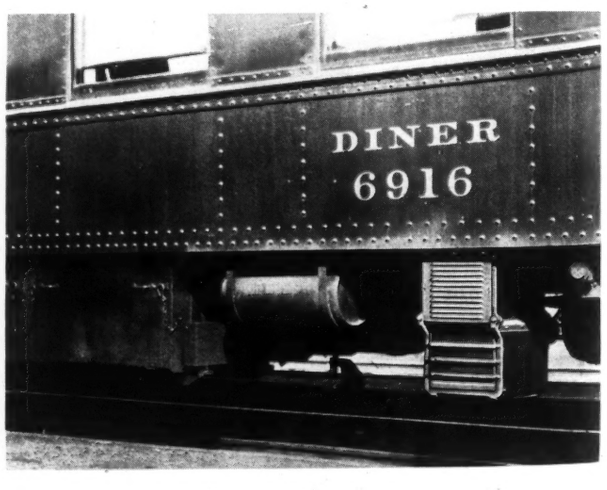
● A multiple-unit system of air conditioning for railway cars recently has been developed. The upper section of each unit contains an air filter, blower and motor and recirculating damper and the lower section contains a cooling and dehumidifying system, condenser and expansion coils. The upper section is a complete air circulating unit in itself, and the lower section is designed to be removed during weather when car cooling is not required.

The units are attached to the car by means of a cast-steel mounting plate which is permanently fastened to the car under-frame. The mounting plate is of a manifold type having ports in the upper side which connect with the discharge and return air ducts in the car and register on the lower side with ports in the conditioning unit. Plates and units are interchangeable.

The illustration at the bottom of this column shows a complete unit (both upper and lower sections) installed on a dining car.

The number of units necessary to use per car vary with the car's size and the conditions to be maintained. It is stated that usually four of the units will be required, each unit conditioning about 12 ft. of interior car length. Each unit weighs approximately 340 lb. Cooling is governed by two thermostats, half of the units on the car being connected with each control; thermostats are set so that one group operates at full capacity before the second group is cut into operation. During the heating season, manually-controlled switches regulate air flow over the heat coils and the regular car thermostat regulates the amount of heating.

Power for operating the units is taken from the car lighting system or from a separate axle-driven generator and battery system; the units are regularly built for



32-volt direct current; a four-unit system requires about 2½ hp.

This method of conditioning railway cars is announced by the Modine Manufacturing Company of Racine, Wis., through its railway sales division at 333 North Michigan avenue, Chicago, Ill.

Electrodes for Welding Aluminum

● An electrode for welding aluminum, designed for either metallic or carbon arc welding, is a five per cent silicon aluminum alloy and can be used for welding sheet or cast aluminum. The electrode is provided with a coating to prevent excessive oxidation and to dissolve any aluminum oxide that may be formed.

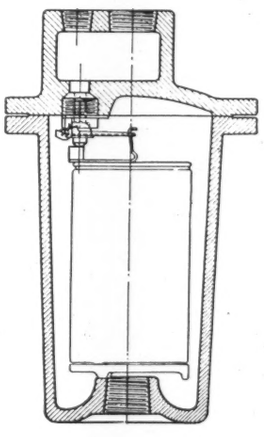
In its use, a short arc of about twenty volts is held with the coating practically in contact with the work. Current will vary with the thickness of sheet or material welded; the following table gives an idea of the proper currents:

Electrode Size	*Current Range*	*Average*
3/32-in.	35-45 amp.	40 amp.
1/8-in	55-80 amp.	70 amp.
5/16-in.	100-150 amp.	125 amp.

The electrodes are produced by the Lincoln Electric Company of Cleveland, Ohio, in four sizes: 3/32, 1/8, 5/32, and 3/16 in. Length is 14 inches.

High Capacity Steam Traps

● High capacity is a feature of an improved line of steam traps lately developed and utilizing the inverted bucket principle. Wearing parts are of stainless steel.



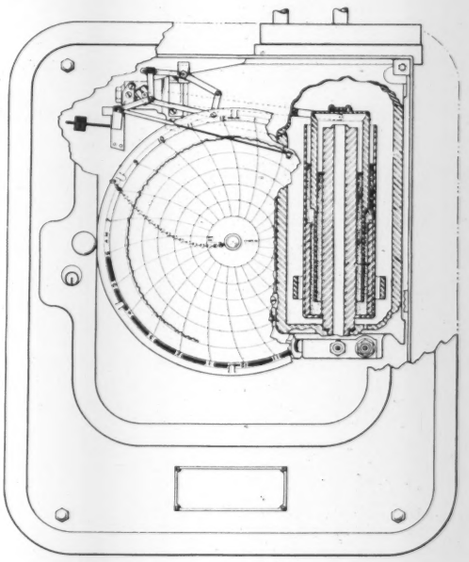
The operation is as follows: A valve lever loops over two guide posts at one end and engages the bucket hook at the other. When the bucket floats, it pushes directly against the valve stem, forcing the valve to its seat. Steam pressure within the trap holds the valve against the seat. Water entering the inverted submerged bucket causes the bucket to lose its buoyancy and exert a pull on the end of the lever. The valve is opened as soon as the weight of the bucket multiplied by the high leverage is greater than the pressure holding the valve against the seat. When opening, the valve lever fulcrums on knife edges which bear directly on the valve seat. The knife edges are designed to reduce friction and to make possible the opening of large-diameter orifices.

These traps are furnished in cast semi-steel for pressures up to 250 lb. In the forged steel pattern, the traps are available for 600, 900 and 1,500 lb. pressure with any degree of superheat.

They are made by the Armstrong Machine Works, of Three Rivers, Mich.

Boiler Water Level Recorder

● Recently, a device for recording boiler water levels was placed on the market. Its design features are as follows: It co-ordinates the feed water flow and the boiler water level records; it shows the true level in the drum instead of the gage glass level; it uses a direct reading chart and it may incorporate alarm devices for high and low levels. It operates on the difference in level obtained by making



two connections to the water column or boiler drum; one connection being made to the water space and the other to the steam space where a constant water head is maintained in the connection by a steam condensing radiator and a reservoir. The variable head or water space connection is applied over a mercury-sealed bell and the constant head connection is applied to the inside of this bell.

As the boiler water level rises or falls, the mercury-sealed bell falls or rises. This motion is transmitted to the recording pen by a forked lever and a spindle which turns in pressure-tight bearings. A reversing linkage is employed between the spindle and recording pen so that the pen moves upward as the water in the boiler drum rises and moves downward as the water level drops.

The water level recorder may be incorporated with a feed water flow meter to record the two related factors on one chart. Remote indicators or recorders operating from the main recorder may be placed at other points. The device is manufactured by the Bailey Meter Co., of Cleveland, Ohio.

Drip Traps of Compact Design

● For keeping risers, steam mains, unit heaters and similar apparatus free of condensate and air, a drip trap has recently been developed which is designed to combine large capacity and compactness. A feature of the trap is a double-seated valve which is balanced against steam pressure; the upper valve closes with steam pressure and the lower